AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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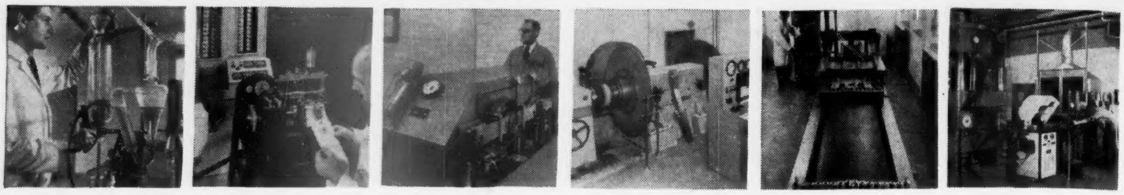
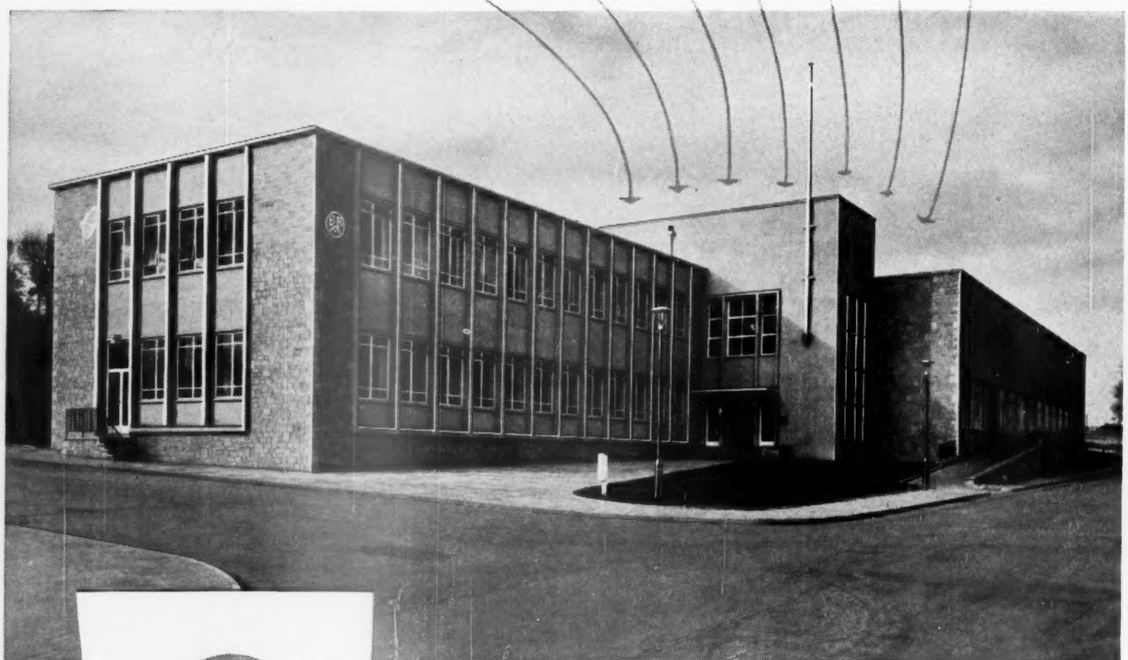


The very heart of the matter



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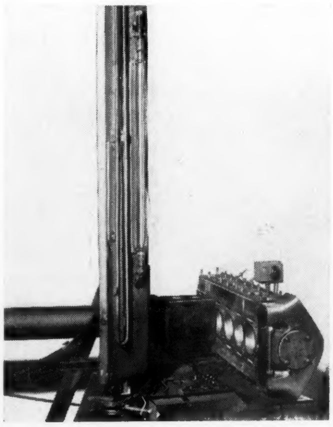
AUTOMOBILE ENGINEER



THE JAGUAR HEAD UNDERGOING DEVELOPMENT ON THE BLOWER RIG DESCRIBED IN THE ARTICLE ON THE WESLAKE LABORATORIES

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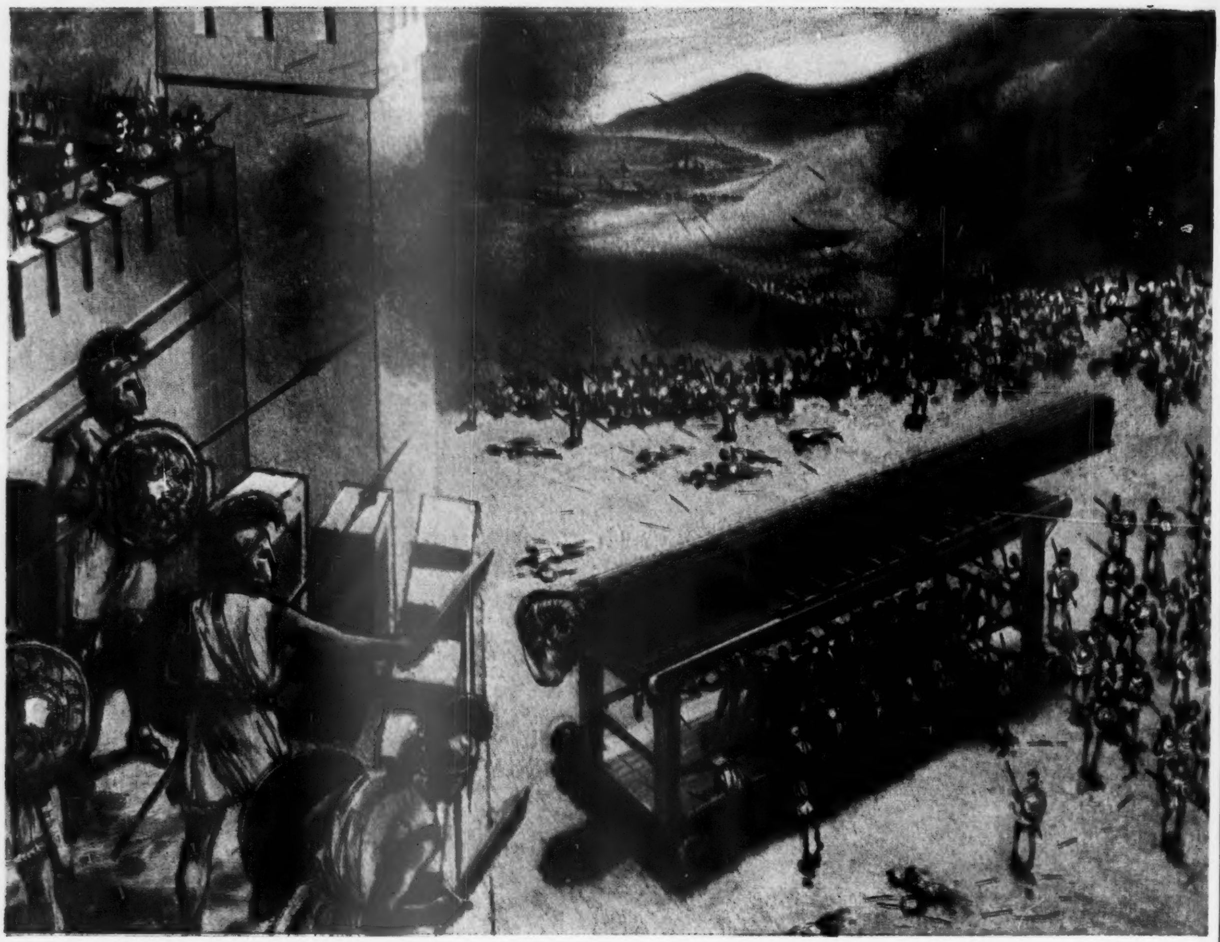


Editor J. B. Duncan

Editorial Staff T. K. Garrett, A.M.I.Mech.E. A.F.R.Ae.S.(*Associate Editor*)

F. C. Sheffield

DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Outlook for 1957

MANY of the current difficulties experienced by the motor industry are obviously only temporary, so it is still necessary to plan largely to meet competition by cutting costs and making the product more attractive and reliable. A great deal has already been done in these directions but, of course, there can be no respite: progress stems only from dissatisfaction with what has already been done.

So far as bodywork is concerned, although styling is not a feature that can be changed frequently, it is as well to remember, when planning new models, that pleasing effects can be obtained by simplicity of line. Good balance and proportioning are essential features in the attainment of simplicity of style. A competent designer, with a good artistic taste and an appreciation of current trends in fashion, neither seeks nor needs to use elaborate chromium plated embellishment for his creations.

One of the most recently developed trends in this country has been the introduction of light vehicles designed specifically for commercial use. This line of approach has many advantages by comparison with the adaptation of private car chassis for this type of application. With the conventional transmission arrangement, that is, a propeller shaft taking the drive from the engine and gearbox at the front to the final drive at the rear, it is difficult to obtain a floor level that is low enough for all purposes.

This raises the question as to whether there would be an adequate market for a front-wheel-drive assembly comprising an engine and transmission unit for this type of vehicle. In view of their existing commitments, the large motor manufacturers would be unlikely to undertake such a project. However, the idea might appeal to smaller manufacturers who have the resources to produce an engine and transmission unit of this type in large numbers, with a view to supplying it to motor manufacturers in general.

The front-wheel-drive layout is, perhaps, particularly well suited to light commercial vehicles, because there is a tendency for the greater part of the payload to be carried on the rear wheels. Such an arrangement would give a good weight distribution in the laden condition, but the rear end would, of course, tend to bounce if the vehicle were lightly laden. In this latter respect, a rear-engine layout is much better, but suffers from the obvious disadvantage that it tends to obstruct the entrance at the back of the vehicle. An extensive investigation would be necessary to determine the most acceptable layout for the engine and transmission. The aim, of course, would inevitably be towards using to the best advantage the space available.

Bearing in mind the fact that there are advantages in giving the driver a seating position higher than that of the passengers, it might be worth while to investigate the practicability of adapting a conventional in-line engine for mounting horizontally under the floor of the driver's cab. There is also much to be said for the employment of a vertical in-line engine, installed transversely under the seats in the cab. However, such an arrangement is extremely difficult to realize, both from the point of view of the layout of the transmission and of lateral weight-distribution; clearance for the steered wheels is a further factor. The more conventional installation of an in-line engine, mounted longitudinally immediately in front of the seats, wastes a lot of space in the cab. The horizontally-opposed type of engine, despite its relatively high cost, should not be discarded without careful consideration. An engine of this type might be installed under the toeboard ramp.

So far as suspension developments in the coming year are concerned, devices for the maintenance of a constant static deflection, regardless of the load carried, are worthy of serious consideration. They are useful both for commercial vehicles and private cars, since they enable the best possible use to be made of the available travel between the bump and rebound limits. Most of the systems currently employed are self-adjusting, but this tends to make them expensive. Possibly a manual adjustment device, for use in conjunction with a simple indicator, might be installed in the vehicle. The simplest arrangement would be a two-position control to adjust for normal and overload conditions.

At first sight, it appears that there is little to be gained by the employment of power-assisted steering on the medium-weight and lighter private cars. However, there are signs that the application of power-assistance may be extended downwards into these ranges. The equipment is likely to be offered as an optional extra, largely for markets in which power-assisted steering is much more common than in this country.

Much of the criticism that has been levelled at the British motor industry recently is unjustified. During the last year, for example, our manufacturers have made considerable advances in the development of new transmission systems for the smaller passenger cars and are undoubtedly ahead of all other countries in this field. British manufacturers are also ahead of their European competitors in power-assisted steering developments, and they have retained their lead throughout the world so far as brakes are concerned. In short, they have no cause to be ashamed of their achievements during the past year.

NON-CURRENT SPARES PRODUCTION

A Sequence-Control System for the Automatic Operation of General-Purpose Machine Tools

H. J. PEARSON

DEMAND for non-current spares will vary very considerably from model to model and will reach a peak value after differing periods of service and spares production must, therefore, be classed as jobbing-work. As such, it requires an approach different from that for a mass production or a quantity-production run. One way is to set up a batch-production shop with general-purpose equipment that will accommodate a variety of work at any given time and this approach is, in fact, being considered by at least two of the major companies.

Batch production presents many problems in speed and economy of manufacture. The use of special machine tools is usually impracticable, owing to the variety of work in progress at any one time, and because of the long setting-up period often required for this type of machine. Although the general-purpose machine is much less efficient than single-purpose equipment for a specific operation, it does permit a range of allied operations to be undertaken with the minimum capital expenditure and floor-space, although it frequently requires additional tooling.

Full use of the available labour is of the greatest importance. On most general-purpose and semi-automatic machines, an operator is required only during a small part of the work-cycle, but the intervals between necessary interventions are usually so short that he has to remain continuously at the machine. Often he merely loads the machine and links together certain automatic or self-actuating phases of a work-cycle, and is prevented from doing any other useful task by the shortness of these phases. If as many automatic functions of a machine-cycle as possible can be continuously linked, manual intervention need occur only at one or two places at the most. The operator then has sufficient time in the intervals to control another machine or to undertake a follow-up operation on the same part.

An automatic programming system that is sufficiently flexible to be applied to short-run production, can obviously be a considerable asset in jobbing-work. An ingenious system of this type for co-ordinating all the functions of a machining cycle and which is at the same time comparatively inexpensive to apply, has been developed by Ian Nickols, Ltd., of Oxted Mill, Oxted, Surrey. This system, which is the subject of a patent, provides a means of bridging the gap between special-purpose, programme-controlled, transfer type machines and general-purpose, semi-automatic, production equipment.

In essentials it is a system for control of workholding, slide-movement, spindle-speeds, feeds and the associated functions that are normally under the manual control of the operator. It is a flexible system, inasmuch that it is suitable equally for the older type of machine, with restricted facilities for automatic operation, and for present-day equipment embodying a number of linking devices for parts of a related cycle. A typical use of it—and one which, from the point of view of versatility, shows it to good advantage—is the automatic operation of a capstan-lathe without the necessity of using plate-cams.

Control elements

It is based upon the use of an electrical multibank stepping-switch which signals—through electropneumatically-operated valves—a pre-set programme to air-cylinders that control the moving elements of the machine and its auxiliary functions. Commercially obtainable elements are used throughout the system. Air-cylinders initiate the movements and the extent of movement is governed by positive stops already existing on the machine. At the completion of a movement, a limit switch is closed by a moving part of the machine and this closure is used to signal the next stage of the work cycle.

One of the advantages of the system is that it can be applied with equal facility to a simple operation requiring the control of only four or five functions, or to an extremely complicated cycle of fifty or more sequence-controlled stages. In each instance, the cost is proportional to the extent of its application and the unattended machine time—or uninterrupted "self-act" time—required.

The method of control was evolved to a large extent by the desire to introduce a means of automatic control for capstan lathes operating on short-run and medium-run work as this appeared to offer the largest number of variables in

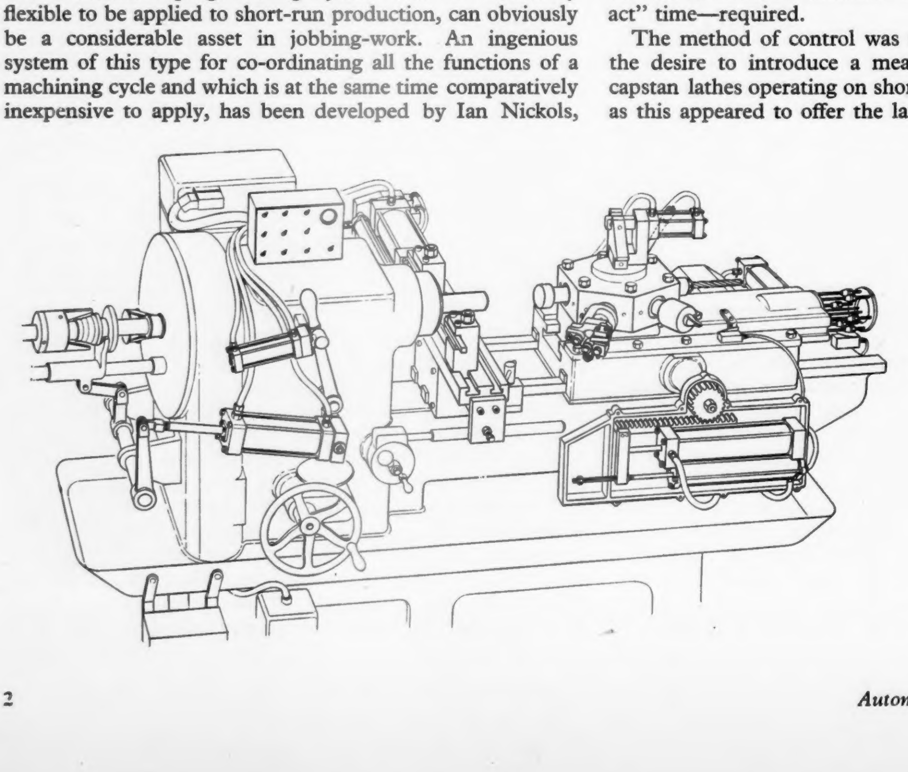
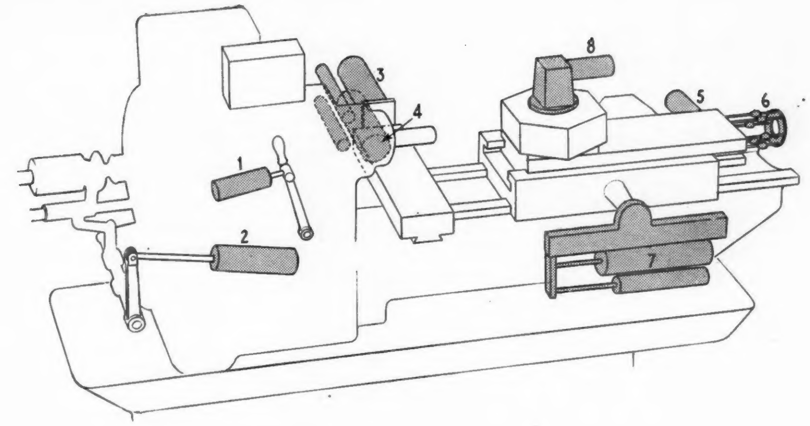


Fig. 1. A Morey 2G capstan lathe fitted with automatic-control units. The functions normally undertaken by an operator are controlled automatically through an electro-air switching circuit which is used in conjunction with a programme-control box. The signalling necessary to indicate that a stage of working has been satisfactorily completed is obtained by the use of simple micro-switches



1—spindle speed control, 2—collet-opening control, 3—rear tool-post actuating-cylinder, 4—front tool-post actuating-cylinder, 5—turret-feed clutch-control, 6—turret-slide micro-switch trips, 7—turret-feed unit, 8—capstan clamping cylinder

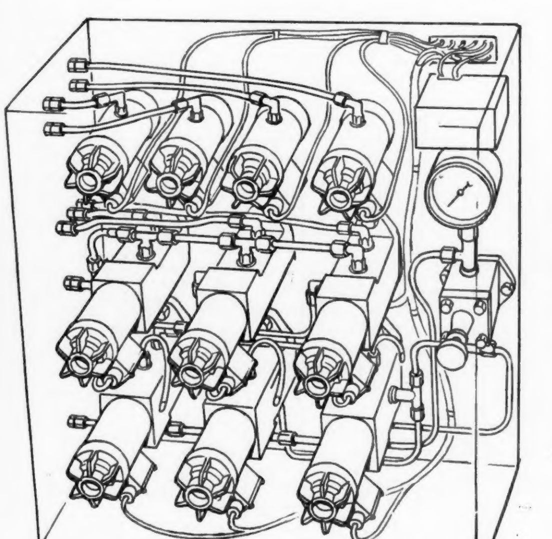
Fig. 2. The location of the main elements of the control system on the modified Morey capstan-lathe

machine-tool operation and a source of substantial practical economy in labour. In this type of application it is possible to maintain the normal capstan-lathe changeover-time of about 2-2½ hr from one component to another, where the additional setting-time for automatic operation varies between 10-40 min, as compared with some 6-7 hr for a conventional type of automatic. Another advantage of this method of operation is that, if after setting-up it is found that the cycle has to be rearranged or tooling changed, it is necessary only to disconnect one set of circuits and to plug in another.

When the system is applied to an existing machine, the control-units are linked together electrically in such a manner that, at the satisfactory conclusion of one part of a cycle, a signal in the form of an impulse, is passed to a control-panel for the start of the next cycle which may consist of any number of actual movements taking place simultaneously, although in a capstan lathe this does not normally exceed five at any one time.

In the capstan-lathe shown in Figs. 1 and 2 the principal functions that need to be linked together for automatic operation are:—

Fig. 3. Arrangement of the Martonair solenoid-operated pilot-valves at the rear of the lathe. These valves are controlled by signals obtained from a multi-bank stepping-switch which governs the sequence



1. Opening and closing of the collet
2. Clamping and unclamping of the turret
3. Feeding and retracting the turret-slide at feed and rapid-return traverses, with dwell periods when facing-tools or form-tools are used
4. Cross-slide movements in two functional directions and return to zero setting; dwell periods when form-tools are used
5. Engaging additional spindle-speeds using both headstock gearing and speed-changing motor
6. Starting, stopping and reversing main motor as may be required for reaming operations, and threading.

These functions are controlled by simple electrical circuits (Figs. 5 and 7), which operate air-cylinders for rapid traverses, and air-cylinders and Hydro-check cylinders for feed control. This equipment is attached to the machine to operate the existing lever-operated and wheel-operated mechanisms. The sequence of the functions is controlled electrically from a control-setting box, such as illustrated in Fig. 4. This box receives a signal from the machine that the preceding phase of the operation has been completed by the closing of micro-switches at each end of a piston movement, and imparts the signal to the uniselector switch that the next part can commence.

Electrical circuit

The basis of the electric circuits for the programming control of a capstan lathe is shown in Fig. 5 and consists of two switching-circuits. The bank on the extreme right—including the customary motor-starting equipment—is normally attached to the machine as part of the additional switchgear and is mounted on the headstock-casing as shown in Fig. 1. The centre circuit is the programme unit and is contained in the control-box already mentioned while the uniselector stepping-switch on the left of the illustration is on the reverse side of the programme-setting panel and is used to co-ordinate the required functions during each phase of an operation.

The signals used during an operation are shown between the setting-panel circuit and machine switchboard. When, for example, the turret is to be fed towards the component, a solenoid-controlled four-way air pilot-valve, such as seen in Fig. 3, is energized to cause the apron-mounted actuating-piston to extend. This movement will continue until an adjustable-position stop on the turret-slide comes into contact with a micro-switch to indicate, in conjunction with the turret-slide positive-stop, that the turret has been advanced the full amount. Completion of this phase of the work cycle is then signalled to the control-box by the micro-switch circuit. When the control-solenoid is released by the next stage of uniselector-switching, the four-way

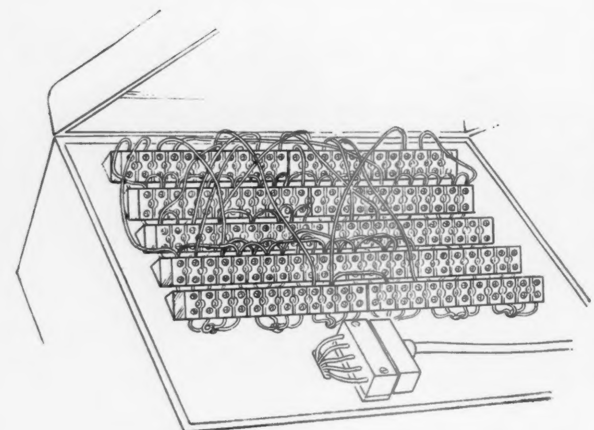


Fig. 4. The programme-control setting-box showing the looping-in of a circuit function from one stage of the cycle to the next. These setting panels are removable and a programme can be built up while another job is in progress. The required circuit is then inserted as a unit

pilot valve allows the turret-operating piston to retract to its full extent in the closed direction. After this movement is completed, the turret-back micro-switch is closed and, in so doing, imparts another signal to the control-box to signify that a satisfactory traverse has been obtained.

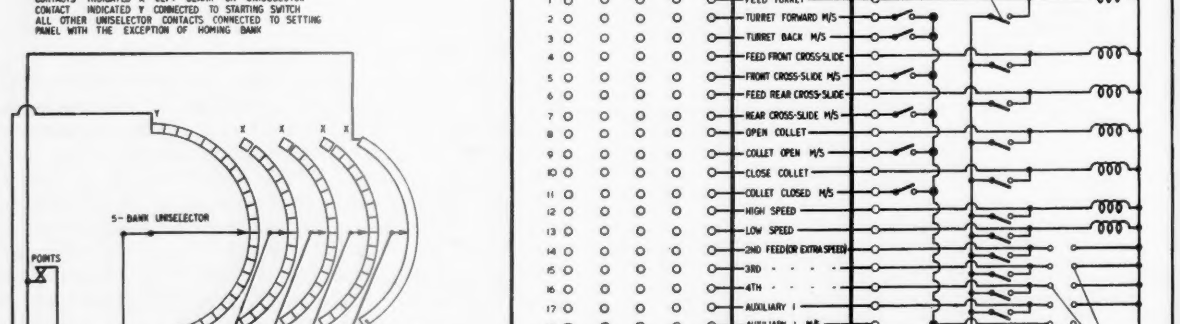
This positive method of signalling, by the use of open, spring-contact type, micro-switches, that a traverse has taken place has been adopted in preference to pressure build-up at the end of a stroke, as it permits a Hydro-check cylinder to be used. Another and equally important reason for micro-switch signalling is that the valve equipment required for sequence-operation through pressure build-up is not sufficiently adaptable or economical when the requirements of the equipment are continually changing.

A similar form of micro-switch signalling is used for controlling the movement of the collet. In the first place, the command for "open collet" is passed to the machine switchboard from the control-box, then through the solenoid-controlled valve shown on the extreme right of Fig. 5. The actuating cylinder, pivotally mounted at the front of the machine, then extends to move the "collet-open" lever, and a micro-switch is closed at the limit of lever-movement to indicate that the opening has taken place. When the collet has to be closed, the solenoid-controlled valve is released, and the main-air circuit causes the piston to retract, and the full extent of movement is signalled by the "collet-closed" micro-switch.

The second lever on the headstock shown in Fig. 2 is used for controlling spindle-speed changes, and is operated in a manner similar to that described for collet operation. When additional speed-changes are required, a speed-changing motor is used with one or more auxiliary sockets in the wiring diagram. The other auxiliary functions in the switchboard circuit can be used for providing a keying-link when special attachments are used. The remaining five connections at the bottom of the setting-panel are used for synchronization, setting, solenoid control, micro-switch control, start and automatic operation to give the energizing link between the mains supply in the machine, starting switchbox, and programme control-unit.

The setting-panel in the centre of Fig. 5 is simply a convenient means of linking together the functions required by a tool layout. In this example, the panel consists of five banks of plug-in sockets, of which the vertical bank on the extreme right provides the direct connection between the

Fig. 5. The basis of the electrical circuit for capstan-lathe operation. It consists of two switching-circuits where the panel on the right shows the additions to the machine switchboard illustrated in Fig. 1. The centre panel is the programme-setting box and illustrates the various functions that can be controlled. These phases of an operation are co-ordinated by the uniselector switch on the left into a series of stages in working



switchboard and the control-box. The actual connection between these two parts is obtained by multicore-cable and multipin-plugs. The remaining banks of sockets are connected to a five-bank, 25-position, impulse-controlled uniselector switch on the left of Fig. 5. The fifth bank is used for the homing signal for synchronization, and the 25th position of the wiping-arm contactor of a uniselector switch is used to signal the first position for the start of a new work-cycle. These banks are arranged horizontally in groups as shown numbered 1-24 on the left, so that up to four circuits can be energized simultaneously when the uniselector-switch contacts are in any one of the 24 positions, giving a possible total of 96 functions in a complete work cycle.

Uniselector-switch control

Leads from the uniselector switch are connected to the setting-panel sockets by multipin-plugs. This arrangement allows a programme to be built up from a layout-sheet for a particular component, ready for insertion into the control box when required, and stored independently of the machine for future use.

A four-bank uniselector switch of the type mentioned, made up from three operating-banks and a homing-bank, is shown in Fig. 6. As in the case of the five-bank type, the actual step-switching is obtained by a solenoid which, in turn, is controlled by the signal from a micro-switch indicating the satisfactory completion of a work cycle. As the solenoid in the uniselector returns after an impulse, a pawl-and-ratchet mechanism moves the multi-bank wiping-arms by one position and closes the circuits for that phase of the operation. In addition to the quick make-and-break switch there is a time-delay switching circuit in the control-box. This circuit is used for incorporating a 5 sec dwell period in a phase of the cycle, such as might be required for bar-feed, form-tools or reaming. When a machine is being set up, the operating-circuits are opened and closed by the manually controlled switches in the headstock-panel and can be selected in any order.

Arrangement of tooling

As an example of the operation of the system in practice, the part shown in Fig. 8 has to be produced using the tool-layout illustrated in Figs. 1 and 9. This layout is arranged in the manner best suited to show the potentialities of the system rather than as a criterion of tool-layout for the operation. It consists of:—

1. Feed to bar-stop
2. Feed roller-box tool

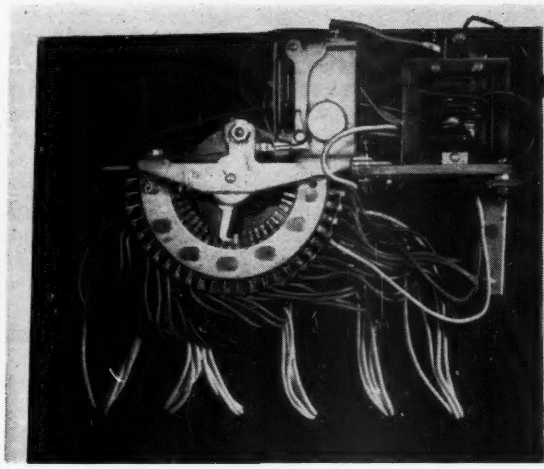


Fig. 6. The rear view of a programme-control panel showing a four-bank uniselector switch. It is a 25-position switch which is indexed by the solenoid on the right receiving an impulse from a micro-switch

3. Centre-drill and face the end of the component
4. Drill and chamfer; bring in form-tool on rear cross-slide
5. Ream
6. Parting-off on front cross-slide.

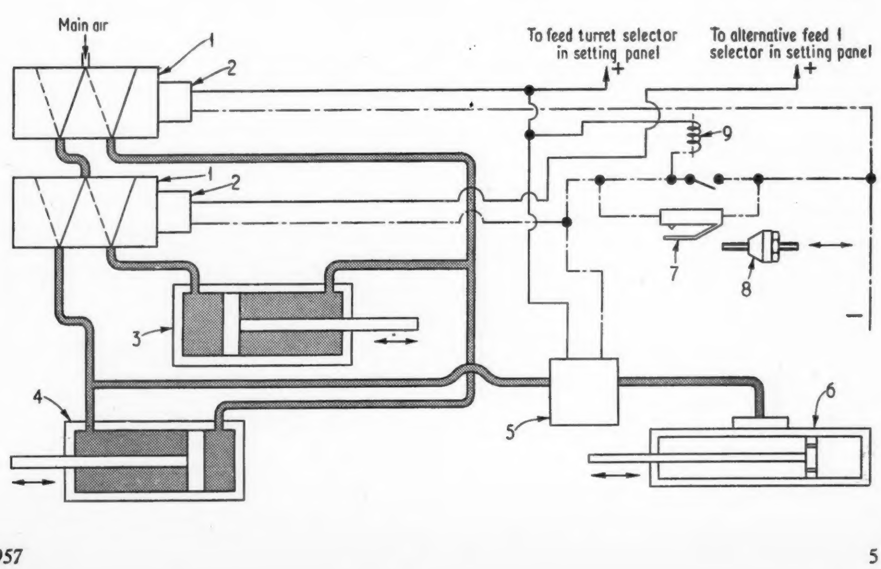
The programme setting to produce these operations is shown in the looped-in wiring-diagram, Fig. 10. The four banks of uniselector contacts are shown on the left, connected in a pattern for sequence operation with the machine-element socket-type connections on the right. A layout of this kind can be prepared from a printed chart and be built up by the planning department and then passed to the shop-foreman or machine-setter to be put into effect.

An analysis of the operation for automatic control shows that there are 16 stages in the work cycle, an additional stage to index the turret for the unused station, and a total of 18 impulses to the uniselector switch to complete the operation. Commencing with the turret in the back position, a bar loaded in the machine (and held in the forward position by gravity-feed) the bar-stop in the turret facing the headstock and the collet closed, the sequence of movements is:—

1. Open the collet and feed the turret with bar-stop
2. Continue feeding the turret and signal "collet-open"
3. Maintain the turret in the forward position against the positive stop and close the collet

1—4-way air-operated pilot-valve, 2—solenoid, 3—machine-feed clutch-control, 4—turret-slide cylinder, 5—solenoid-operated valve, 6—turret-slide Hydro-check cylinder, 7—micro-switch for turret-forward stop, 8—cam-trip on turret-slide, 9—latching-relay

Fig. 7. Diagrammatic layout of the turret-feed circuits. This allows rapid-traverse, machine-feed or Hydro-check feed to be used



4. Retract turret for indexing
5. Feed turret with roller-box tool, using 2nd feed (the 2nd feed will be explained at a later stage)
6. Retract turret for indexing
7. Feed turret with combination centre-drill and facing cutter, using 2nd feed; bring time-delay at full depth of cut before retracting
8. Retract turret for indexing
9. Feed turret with combination-drill and shell-type chamfer cutter, using 2nd feed; feed in rear cross-slide
10. Continue feeding turret with 2nd feed; continue feeding cross-slide and bring in time delay at full depth of forming before retracting
11. Continue feeding turret with 2nd feed until contact is made with micro-switch and positive stop
12. Retract the turret for indexing and simultaneously engage low spindle-speed
13. Feed the turret with reamer and simultaneously engage normal machine feed (normal feed engaged by retracting the piston controlling the clutch and does not require a plug-in the panel)
14. Maintain the turret in the forward position; stop the spindle motor; bring in time-delay while reamer is being withdrawn
15. Retract the turret for indexing and engage high-speed while spindle is stationary
16. Start motor; part-off workpiece using the front tool-post on the cross-slide; retract turret for indexing
17. Retract turret for indexing to start position
18. Home.

This programme is built-up by taking the open-collet lead, feed-turret lead, and turret-forward micro-switch lead from the connecting bank on the right of Fig. 10 and plugging-in these into sockets in the first horizontal row of the four-bank uniselector switch contacts, so that the two operations start simultaneously through the medium of the solenoid-controlled pilot valves. As the turret-slide reaches the end of its forward travel it closes a micro-switch by means of the drum-stop shown in Fig. 1 and comes to rest against the positive stop beneath the slide. Closure of the micro-switch then passes an impulse to the uniselector-switch solenoid and the wiper-arms are moved to the 2nd position.

As the 2nd stage requires the collet to remain open, and the turret held up by pressure-air against the stop, these two leads from the switchboard are looped in by wander-plug connection from the first row in the manner shown. Completion of this phase of the operation is indicated by the "collet-open" micro-switch. At the 3rd stage the turret has still to be held up against the stop and this lead is looped in from the 2nd row and the "collet-closed" lead and collet-closed micro-switch lead brought in from the switchboard.

At this stage the stock has been advanced by gravity-feed in the normal manner and the collet closed. The solenoid-operated control-valve is then reversed to allow the turret

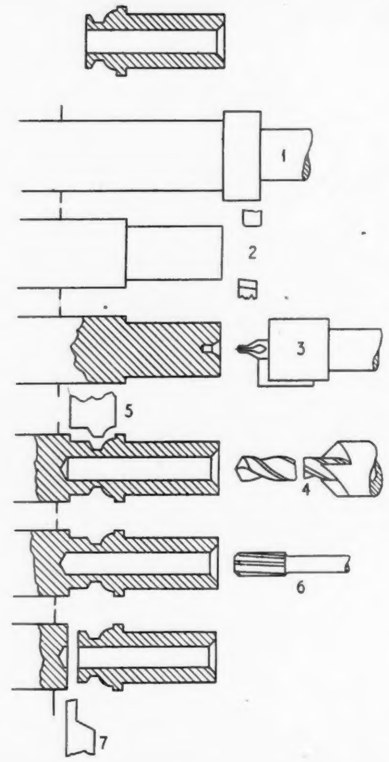


Fig. 8. The sequence of operations for the component discussed in the text. Reading from top to bottom: the finished part; bar feeding; roller-box turning; centring and facing; drilling; chamfering and forming; reaming and parting-off are illustrated in turn

to retract, and a signal for this occurring is obtained by looping-in the turret-back micro-switch lead into the 4th horizontal bank, which is the only function taking place at that instant. The turret is clamped and unclamped automatically by the air-cylinder seen in Fig. 1, the circuit of which is arranged so that it is clamped or unclamped according to direction of movement.

On receiving an impulse that the turret is back and automatically indexed by the machine, the 5th bank of uniselector contacts become energized. As the turret has to be fed forward again, another lead is looped in from the 3rd stage when it was last used, and the "turret-forward" micro-switch circuit-lead is looped in from the 1st stage. The 2nd feed is now used for the first time (in the machine illustrated, for convenience in operation, the 2nd feed is, in fact, the machine feed and the 1st feed is by air-operated cylinder in conjunction with an hydraulic check-cylinder)

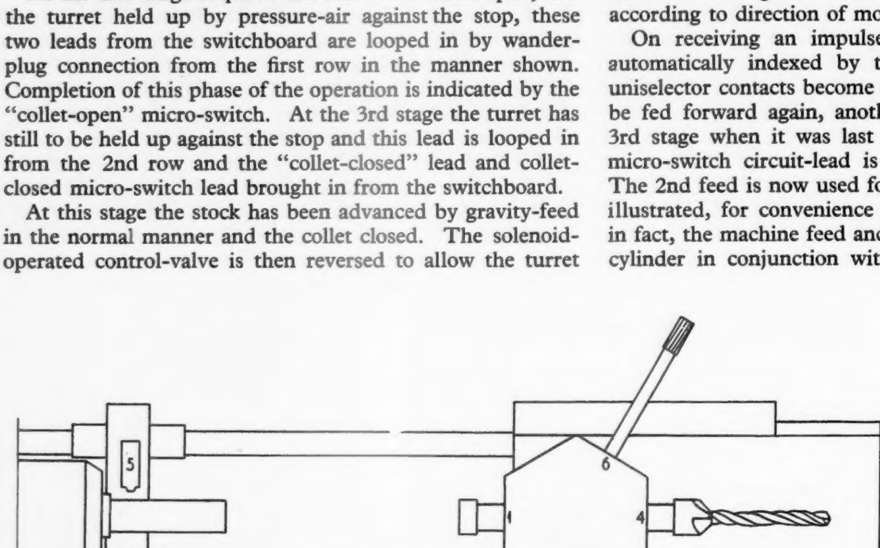


Fig. 9. The tooling layout for producing the component shown detailed in Fig. 8

and a lead is fed in from the connecting bank on the right.

With the signal from the "turret-forward" micro-switch that the roller-box tool has completed its operation, the uniselector switch is indexed to the 6th stage in accordance with the analysis. As the turret has to be retracted for indexing and no other function need take place during this momentary phase, the only looping-in wiring is for the turret-back micro-switch from the 4th stage when this signal was last used.

This practice of looping-in a lead to the next stage once it has been used is a characteristic of the Nickols system and allows a sequence of operations to be assembled in a few minutes once the cycle has been determined. During the 7th stage, the turret is fed with the combination centre-drill and facing-cutter using the 2nd feed and, on coming into contact with the positive-stop and micro-switch contact, is held in the forward position by a 5 sec time-delay to obtain the desired finish on the end of the part. This time-delay is a simple mercury-type switch and the lead is brought in to one of the plug-in sockets of the 7th stage.

Time-delay circuit

The 9th, 10th and 11th stages of the operation, where it is necesary to drill, chamfer, and use the cross-slide for a forming operation serve to illustrate the flexibility of the system. It is, for example, obvious that the drilling operation will require a longer time to complete than the comparatively shallow forming operation undertaken with the rear toolpost on the cross-slide, while on the other hand the forming operation will require a certain amount of dwell-period when the tool is fed to full depth.

Starting with the uniselector contacts set for the 9th stage, the "feed-turret" lead is looped-in from the 7th stage, the "2nd feed" lead also from the 7th stage, while the "feed rear cross-slide" and the "rear cross-slide forward position" micro-switch leads are taken from the connecting bank for the first time. As soon as the rear cross-slide micro-switch signals that the full depth of cut has been reached (this will, of course, occur while the drilling operation is still taking place) the uniselector switch indexes to the 10th position to continue other movements of this phase.

Here, the turret is kept moving in the forward direction by the 2nd feed and the cross-slide is held up by the looping-in, while the 5 sec time-delay switch is brought in from the 7th bank and tripped for holding the cross-slide for the dwell period after which it will be retracted to the null position when the solenoid in the pilot valve is released. When the cross-slide is back, and signalled as being back by a micro-switch, the uniselector stepping-contacts are moved to the 11th position to allow the completion of the drilling and chamfering.

During the 12th stage, when the turret is being retracted after drilling to depth, the low spindle-speed is engaged, in preparation for reaming, by plugging-in the low-speed lead which causes the lever on the headstock casing to be moved. As the turret is being fed with the reamer, the normal feed of the machine is engaged by leaving out the 2nd feed looping-in. When the full extent of forward traverse is reached, the 14th stage is signalled. Here, the turret is held in the forward position, the spindle-motor is stopped, and the time-delay circuit is looped-in from the 10th stage in order that the reamer can be withdrawn while the spindle is stationary.

The 15th stage consists of retracting the turret and engaging the high-speed drive while the spindle is stationary. When the turret-back micro-switch has closed, the uniselector indexes the 16th stage which consists of starting the motor, feeding the front tool-post with the parting-off tool, until contact is made with a micro-switch, and feeding the turret for an indexing motion. Turret-movement is restricted to the minimum necessary for indexing, and the turret is retracted during the 17th stage of the cycle. When the micro-switch is closed at the conclusion of the parting-off stage the uniselector is caused to short-circuit the unwanted 18th to the 24th positions by straight looping-in.

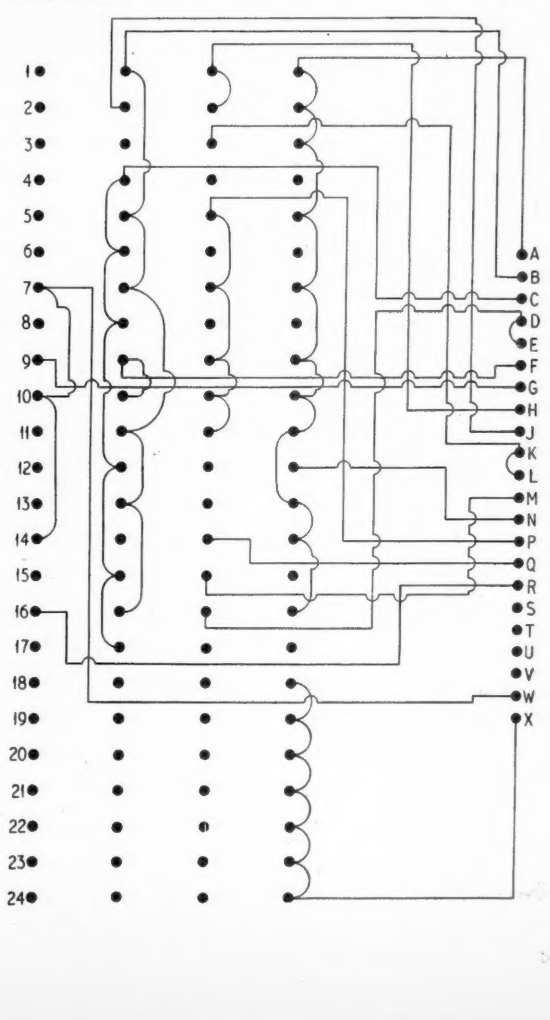
Turret-slide control

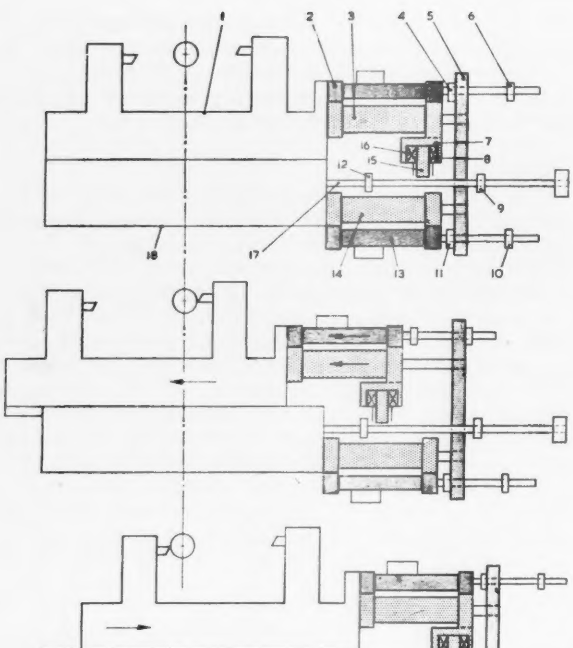
The essence of the system is the retention of the normal controls existing on the machine and the addition of actuating equipment to allow them to function automatically. Basic units have been developed which are essentially attachments in themselves but once installed, become an integral part of the machine. Control equipment for the turret-slide, (Fig. 1) takes the form of a self-contained unit which can fit over the operating shaft of the normal capstan handle. It is a rack-and-pinion mechanism operated by the air-cylinder beneath it, and the Hydro-check cylinder when used for a cutting-traverse. The pinion is attached to the capstan drive-shaft and is rotated by the movement of the rack. The rack, in turn, is controlled by the plate-attachment fitted to the end of the piston of a Martonair* double-acting cylinder. In order that the air-cylinder can be used for both feed and rapid traverse movements, a check unit is fitted beneath

**Martonair Ltd., Parkshot, Richmond, Surrey.*

Fig. 10. The looped-in wiring diagram for producing the component discussed in the text. The control functions on the right connect with the machine switchboard, while the 24 horizontal banks of four sockets are in circuit with a uniselector switch on the underside of the panel

A—feed turret, B—turret-forward micro-switch, C—turret-back micro-switch, D—feed front cross-slide, E—front cross-slide micro-switch, F—feed rear cross-slide, G—rear cross-slide micro-switch, H—open collet, J—collet-open micro-switch, K—close-collet, L—collet-closed micro-switch, M—high speed, N—low speed, P—2nd feed, Q—stop motor, R—start motor, S—1st auxiliary, T—1st auxiliary micro-switch, U—2nd auxiliary, V—2nd auxiliary micro-switch, W—time-delay switch, X—home





1—cross-slide, 2—rear-tool-post Hydro-check, 3—rear-tool-post control-cylinder, 4—Hydro-check resetting-stop for rear tool-post, 5—thrust connection-plate, 6—Hydro-check intervention setting-stop for rear tool-post, 7—extension bracket on control-cylinder, 8—micro-switch for inward movement of front tool-post, 9—positive stop for inward movement of front tool-post, 10—Hydro-check intervention setting-stop for front tool-post, 11—Hydro-check resetting-stop for front tool-post, 12—Positive-stop for inward movement of rear tool-post, 13—front tool-post control-cylinder, 14—front tool-post Hydro-check, 15—positive-stop contact block, 16—micro-switch for inward movement of rear tool-post, 17—positive-stop setting rod, 18—saddle

Fig. 11. Schematic arrangement of control equipment for operating the cross-slide with a positive stop in two functional directions. In the upper diagram the slide is in the at-rest null position; the centre diagram shows the rear tool being advanced by the extension of the upper control-cylinder; in the lower diagram the front tool is being fed towards the work by the extension of the bottom control-piston

the main actuating cylinder, and is arranged in such a manner that the main cylinder functions at its normal rate of traverse until restricted by the action of the check unit.

This item is a British Bellows Hydro-check cylinder† which has a piston with oil on both sides. Oil is displaced from one side of the piston and transferred to the other side as it is moved. The rate of oil flow is controllable by an adjustable restrictor-valve to govern the rate of movement for a given load.

The ram of the Hydro-check cylinder has a threaded extension and, for turret feeding, is coupled directly to the rack drive. After the drive-plate has begun to move away in fast traverse (operated by the main cylinder off the shop air-line) it comes into contact with a stop-nut on the check unit. A unit of this type is shown in Fig. 11. When the turret is right up, and completion of movement is signalled by the micro-switch illustrated in Fig. 7, the Hydro-check piston-valve in the cylinder is opened in order that the hydraulic piston can be retracted with the minimum of oil disturbance. The piston is then reset by the nut on the opposite side of the plate in readiness for the next forward movement.

The extent of turret movement in the forward direction is governed by index stop-pins beneath the slide in the normal manner of a capstan lathe, and the indication that full travel has taken place signalled by leaf-type micro-switches.

†*Benton and Stone Ltd., Bracebridge Street, Birmingham, 6.*

This signalling is obtained by a set of turret-mounted cam-trips mounted at the end of the turret-slide and contained in the same carrier plate as that used for the positive stops. These are on a larger pitch-circle diameter than the positive stops for turret movenent so that there is a double row of six stops in each bank.

These cam-trips for closing the micro-switch consist of screwed rods having a support-plate at their outer end to form a cage. The cam-trips consist of a cylindrical coned nut and a rearward-facing locknut. As the turret-slide advances, carrying with it the indexing stop-pins, and comes into contact with the stop on the machine the coned end of the appropriate trip closes the micro-switch fitted to the end of the lathe bed. The "turret-back" micro-switch is fitted at the front of the saddle and is closed by the plate shown attached to the top of the turret-slide.

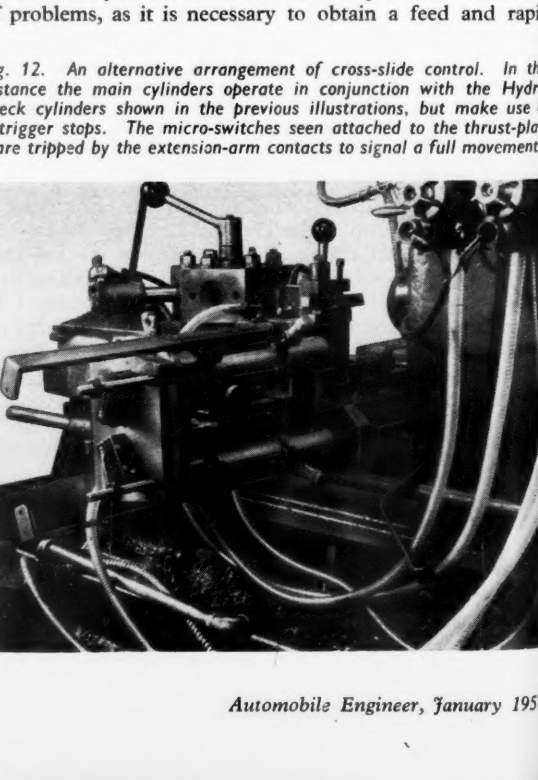
The basis of the turret-feed circuit is shown in Fig. 7, which also shows the interconnection between the three rates of traverse that can be used in any one set-up. They are: fast traverse from the main cylinder; machine feed engagement by the clutch-control cylinder on the left; combined main cylinder and Hydro-check feed by the control cylinder on the right. These feeds are obtained in conjunction with Martonair solenoid-controlled pilot valves and negative-pole electrical switching-circuit. The Hydro-check circuit operates in conjunction with the main air pilot valve and the solenoid-controlled restrictor-valve for applying the check during the forward movement of the hydraulic piston.

It will be seen that the micro-switch used for signalling the "turret-up" also closes the circuit for a hold-in type of latching-relay for controlling the position of the solenoid pilot valve governing the bypass setting in the Hydro-check. In the example discussed, the Hydro-check feed was used for most of the operations while the machine-feed was used for the reaming operation by keeping the clutch-actuating cylinder, seen at the back of the machine in Fig. 2, closed for the drilling operations, and extending the piston during reaming. Under these conditions the cam-trip for the reaming-stage passes a signal to the four-way pilot valve and keeps the Hydro-check bypass valve open during the turret-forward traverse.

Cross-slide control

Satisfactory control of the cross-slide presented a number of problems, as it is necessary to obtain a feed and rapid

Fig. 12. An alternative arrangement of cross-slide control. In this instance the main cylinders operate in conjunction with the Hydro-check cylinders shown in the previous illustrations, but make use of outrigger stops. The micro-switches seen attached to the thrust-plate are tripped by the extension-arm contacts to signal a full movement



traverse in two directions with the slide returning to the null position at the completion of a cycle. This difficulty has now been satisfactorily overcome by the method shown diagrammatically in Fig. 11.

In the first diagram, the cross-slide is in the midway position; in the second, the rear toolpost is advanced; and finally, the front toolpost is advanced. The general problem was that of obtaining a precise movement in the direction of feed and obtaining a micro-switch signal that the movement had occurred. It has been resolved by using two Martonair double-acting cylinders (*3* and *14*) for the movements. The cylinder for controlling the rear toolpost is attached to the slide, and the other to the saddle of the machine, while the pistons of both cylinders are attached to a common thrust-plate, *5*.

When the rear tool is advanced, the thrust-plate (*5*) remains stationary under the restraint of the lower cylinder, *14*, while the operating cylinder, *3*, moves to the left, pushing the slide with it. When the front tool is used, the piston of *14* extends, and transmits the thrust through the retracted piston of cylinder, *3*, and is so doing pulls the slide towards the back of the machine. That is, the pistons in both cylinders are under load during a movement of the slide, and one is used as the transmission-link for the other by introducing air into opposing ends of the paired-operation units which function in conjunction with Hydro-check cylinders.

The Hydro-check cylinders, positive-stops, and micro-switches are arranged as follows: when the rear tool is to be used, as illustrated in the centre-diagram, the actuating cylinder moves to the left, and carries with it the Hydro-check cylinder, *2*. The screwed extension-ram of the Hydro-check passes through a clearance hole in the thrust-plate, *5*. On both sides of the thrust-plate are adjustable-position stop-nuts, *4* and *6*. The slide starts to move in rapid traverse to the left until the ram-extension stop-nut, *6*, comes into contact with the thrust-plate and so checks the feed until the full traverse has been made. On the return-stroke, the Hydro-check piston is reset by stop *4*, while the positive stop for slide-movement is obtained from the bracket *7*, carrying the contact block, *15*, attached to the upper cylinder.

As the actuating-cylinder moves to the left carrying with it, *15*, it comes into contact ultimately with the adjustable stop, *12*, which is a nut on screwed-rod, *17*, attached to the saddle and to an outrigger support at the other end. On the immediate left and right of the contact-block, *15*, are micro-switches, shown *8* and *16*. These switches are operated by a simple lever-mechanism, which is tripped immediately before the positive stops come into contact. The same procedure is used for the front tool where the lower cylinder is the prime mover and is controlled by a positive stop operating in the opposite direction. The switchgear arrangements for starting, stopping and speed-control are, of course, dependent upon the type of starting-switch in use

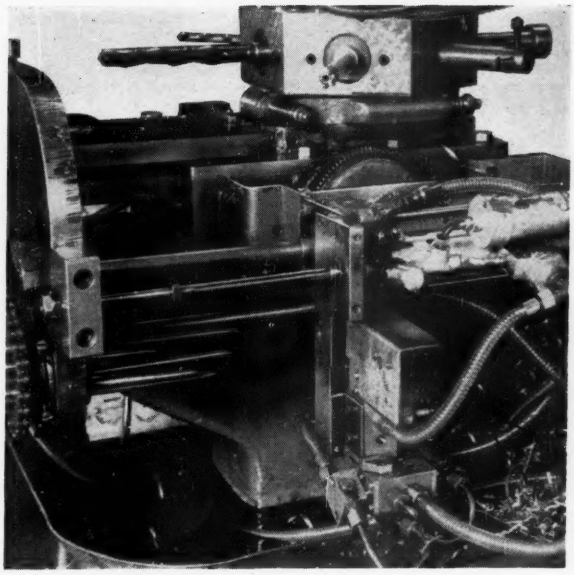


Fig. 13. Another method of controlling the intervention of the Hydro-check cylinder for turret movement. Here, the Hydro-check control-valve is operated by a turret arrangement of stop-pins shown on the left which are indexed by chain-drive from the turret-slide stops

and various types of circuits can be used for looping into the control-box sequence.

The arrangement of the control-elements permits a comprehensive programme to be built up, if necessary, as an operation proceeds without the necessity of pre-planning of feeds, speeds and tool overlap. As mentioned earlier, the essential characteristic of the system is the use of micro-switches at each end of a slide or lever movement to initiate a signal and use the impulse obtained to move the stepping-switch contacts to the next position. Although it may be necessary to use a number of these switches in a comprehensive layout, they cost only a few shillings each. They can, therefore, be regarded as expendable in the event of a breakdown or failure through damage.

The principal advantages of this method of working are, the simplicity of changing over from one workpiece to another, and the relatively small capital expenditure entailed in the provision of automatic control. It can, however, lead to a somewhat complex installation of pipes and wiring to existing equipment that requires careful arrangement and codification to avoid confusion. While a capstan-lathe permits good use to be made of uninterrupted cutting-time, the system can, of course, be applied to almost any machine to obtain the advantages of automatic control.

Aluminium-tin Bearings

ENCOURAGING results have been obtained by the Tin Research Institute from tests conducted in Italy with aluminium-tin bearings of three different compositions, one of which contained titanium. Casting the bearing material containing titanium proved difficult and the final content of this metal was not easy to control owing to segregation and varying losses of the alloying element during casting. An average titanium content was 0·88 per cent.

Best results were obtained from a special alloy, developed by the Institute. The analysis of this alloy is: tin 21·1 per cent, copper 1·83 per cent, silicon 0·15 per cent, iron 0·13 per cent, and the remainder aluminium. This alloy has an ultimate tensile strength of 6·9 tons/in^2, an elongation of 4·8 per cent and a Brinell hardness of 43.

In the tests, it was used in conjunction with an unhardened mild steel shaft, having a Brinell hardness of 124. A wear of 0·0079 in was recorded, this figure remaining stable after 45,000 km running. No further wear had been found to occur after 102,000 km. The rubbing velocity was 6·55 ft/sec. The freedom of scoring of the shaft was thought to be attributable to the large proportion of tin in the bearing. This would allow some absorption of small abrasive particles, which would tend to embed below the surface of the bearing.

*EQUALIZING TYPES OF SUSPENSION**

General Analysis of Layouts of the Systems Employed on the Packard and Citroën 2CV Cars

Dr. Sc. Tech. J. M. Pevsner

DURING the last five years, considerable interest has been shown in suspension systems incorporating a flexible interconnecting-link between the front and rear wheel assemblies. Since suspensions of this type tend to reduce pitching, the Author terms then equalizing suspensions. They differ from most other types in that they reduce the differential between the vertical loads imposed, on the vehicle, by the front and rear axles. Although the basic principle was established many years ago, suspensions of this type were first applied on the Citroën 2CV in 1949, and it was not until 1955 that an equalizing suspension was introduced on the Packard cars.

General Principles

One of the simplest layouts of this type is illustrated in Fig. 1. Interposed between the body and the front and rear wheel assemblies on each side of the vehicle are elastic elements having stiffnesses $C_{1\delta}$ and $C_{2\delta}$ respectively. An equalizing beam on each side couples the front wheel assembly to the rear one. Each end of this beam bears on an

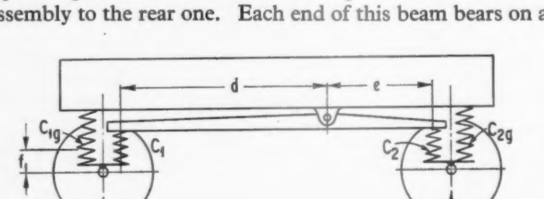


Fig. 1 Diagrammatic illustration of equalizing type suspension

additional elastic element interposed between it and the adjacent wheel assembly. A pivot, at a distance d from the front end and e from the rear end, secures the beam to the frame of the vehicle. As can be seen from the illustration, the stiffness of the front equalizing spring-element is C_1 and that of the rear one is C_2.

Assume that the front wheels have been displaced by an amount f_1 relative to the body. The reaction P_1 on the front wheels will have been increased because of the compression of the springs C_1 and $C_{1\delta}$. In these circumstances, the reaction due to the compression of C_1 is transferred by the equalizer to C_2 and, therefore, increases also the reaction at the rear wheels. Similarly, displacement of the rear wheels by an amount f_2 increases the reaction at the front wheels. If the springs have a linear characteristic, then generally:

$$\left.\begin{aligned} P_1 &= C_{11}f_1 + C_{12}f_2 \\ P_2 &= C_{22}f_2 + C_{21}f_1 \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where C_{11}, C_{12}, C_{22} and C_{21} are the relative stiffness values. C_{11} and C_{22} can be called the basic stiffnesses of the front and rear suspensions, whilst C_{12} and C_{21} are the stiffnesses of the connecting system.

To determine the values of C_{11} and C_{21}, the body and the rear wheels are assumed to be at a constant height relative to the ground, that is, $f_2=0$, while f_1 is the displacement of the front wheels. Then:

$$P_1 = C_{1\delta} f_1 + C_1(f_1 - \alpha d)$$
$$P_2 = C_2 \alpha e$$

where α is the angle of rotation of the equalizer. If the equalizer is to remain in equilibrium:

$$C_1(f_1 - \alpha d)d = C_2 \alpha e^2$$

Eliminating the angle α from these equations:

$$P_1 = f_1\left(C_{1\delta} + \frac{C_1C_2e^2}{C_1d^2 + C_2e^2}\right)$$
$$P_2 = f_1\frac{C_1C_2de}{C_1d^2 + C_2e^2}$$

If $f=0$, equations (1) become:

$$P_1 = C_{11}f_1 \text{ and } P_2 = C_{21}f_1$$

These values can be substituted in the preceding pair of equations to give:

$$C_{11} = C_{1\delta} + \frac{C_1C_2e^2}{C_1d^2 + C_2e^2}$$
$$\text{and } C_{21} = \frac{C_1C_2de}{C_1d^2 + C_2e^2}$$

Similarly, if $f_1=0$, and assuming the displacement of the rear wheels to be f_2:

$$C_{22} = C_{2\delta} + \frac{C_1C_2d^2}{C_1d^2 + C_2e^2}$$
$$\text{and } C_{12} = \frac{C_1C_2de}{C_1d^2 + C_2e^2}$$

that is, $C_{21} = C_{12}$.

The fact that these two are equal is explained by the well-known theorem of relative displacement. For all equalizing suspensions incorporating springs with linear characteristics, C_{21} is always equal to C_{12}, although the actual designs may differ and the stiffnesses of the elastic elements might be introduced in various combinations into the expressions of C_{11}, C_{22}, C_{12} and C_{21}. The values of C_{11}, C_{22}, C_{12} and C_{21} for any layout of this type can be determined in this way.

To find the values of static deflections f_{01} and f_{02} respectively of the front and rear suspension, it is assumed that in equation (1) $P_1 = G_1$ and $P_2 = G_2$, where G_1 and G_2 are the static loads at the front and rear wheels. Then:

$$G_1 = C_{11}f_{01} + C_{12}f_{02} \text{ and } G_2 = C_{22}f_{02} + C_{21}f_{01}$$

and consequently:

$$\left.\begin{aligned} f_{01} &= \frac{C_{22}G_1 - C_{12}G_2}{C_{11}C_{22} - C_{12}C_{21}} \\ f_{02} &= \frac{C_{11}G_2 - C_{21}G_1}{C_{11}C_{22} - C_{12}C_{21}} \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

Thus, the static deflection at each end depends upon the load imposed both on the front and on the rear wheels.

Design

In Fig. 2, the equalizing suspension of the Citroën 2CV is shown. The front and rear wheels are carried by bell-crank levers pivoted on the frame in such a way that they move in vertical planes. Pendant ends of these levers are connected to the suspension springs. There are two spring assemblies, one on each side, and each is common to a front and a rear wheel suspension lever. Fig. 3 shows the spring arrangement. Tension

**Translated from Avtomobilnaia i Tractornaia Promishlenost, No. 3, 1956.*

Fig. 2. On the Citroën 2CV, a single spring-assembly is mounted on each side of the chassis and it serves both the front and rear wheels. It is connected to the short arms of the bell crank levers, on which the wheels are mounted

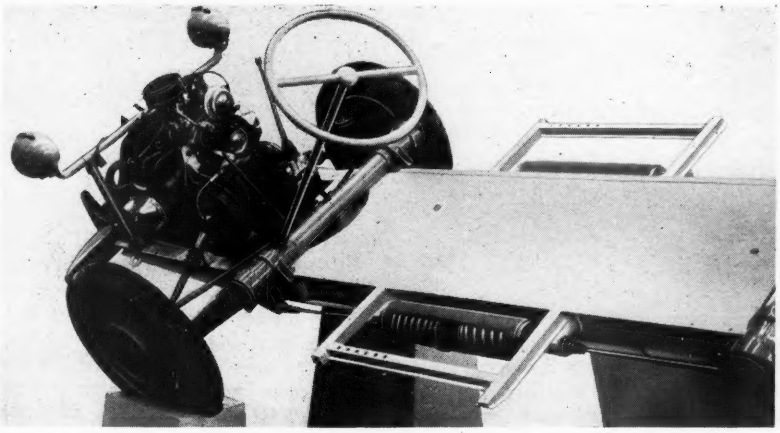
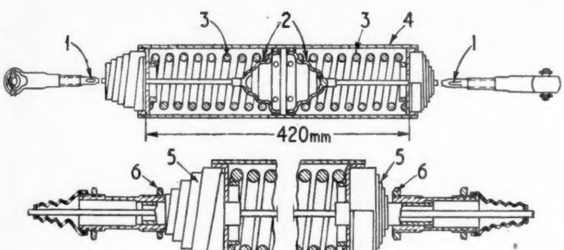


Fig. 3. The Citroën 2CV spring assembly, showing the two coil springs of the main suspension system and the volute springs that stabilize the vehicle



rods 1, connected to the pendant levers, transmit the forces through their dished ends 2, to the two coil springs 3, which are loaded in compression. The springs are carried in a common housing 4. To ensure stability, additional elements are used, in the form of auxiliary volute springs 5, and they are inserted between the ends of the housing 4, and guides 6, mounted on the frame.

The suspension has a low stiffness value so far as pitching is concerned. Thus, if the front wheel moves upwards relative to the frame, while the rear one moves downwards, the spring 3 does not oppose the motion. This makes it possible to reduce both the pitching frequency and the angular acceleration. However, this low angular stiffness also reduces the resistance to angular displacement when the car is accelerated or braked.

To determine the stiffness of the suspension layout, Fig. 4, it is assumed that the displacement of the rear wheels is $f_2=0$, and that of the front wheels is f_1. The resultant reaction at the front wheels is:

$$P_1=\left(f_1\frac{l'_1}{l_1}-y\right)C_1\frac{l'_1}{l_1}$$

$$=f_1C_1\left(\frac{l'_1}{l_1}\right)^2-yC'_1\frac{l'_1}{l_1}$$

where l_1 and l'_1 are the effective lengths of the wheel arms, C_1 and C'_1 the stiffnesses of the main and auxiliary springs respectively, and y is the displacement of the spring housing. The force C_1 exercised by the spring upon the housing is counteracted by the main spring C_2 and the auxiliaries C'_1 and C'_2:

$$\left(f_1\frac{l'_1}{l_4}-y\right)C_1=C_2y+(C'_1+C'_2)y$$

where the auxiliary springs are pre-compressed and are engaged simultaneously. Eliminating y:

$$P_1=f_1\left(\frac{l'_1}{l_1}\right)^2\frac{C_1(C_2+C'_1+C'_2)}{C_1+C_2+C'_1+C'_2}$$

and with reference to equation (1):

$$C_{11}=\left(\frac{l'_1}{l_1}\right)^2\frac{C_1(C_2+C'_1+C'_2)}{C_1+C_2+C'_1+C'_2}$$

The reaction at the rear wheels is:

$$P_2=C_2\,y\frac{l'_2}{l_2}=f_1\frac{l'_1}{l_1}\,\frac{l'_2}{l_2}\,\frac{C_1C_2}{(C_1+C_2+C'_1+C'_2)}$$

so that:

$$C_{21}=\frac{l_1'l_2'C_1C_2}{l_1l_2\,(C_1+C_2+C_1'+C_2')}$$

Similarly:

$$C_{22}=\left(\frac{l_2'}{l_2}\right)^2\frac{C_2(C_1+C_1'+C_2')}{C_1+C_2+C_1'+C_2'}$$

$$C_{12}=\frac{l_1'l_2'\;C_1C_2}{l_1l_2\,(C_1+C_2+C_1'+C_2')}=C_{21}$$

Thus, if the values of C_{11}, C_{22}, C_{12} and C_{21} are known, the static deflections of the front and rear suspension can be determined, in accordance with equation (2).

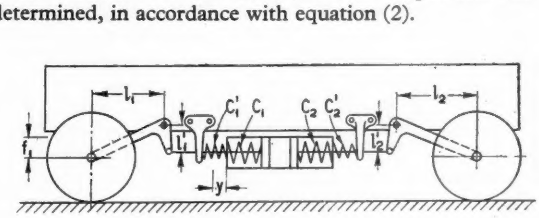


Fig. 4. Diagrammatic layout of the Citroën 2CV suspension

The actual design is more complicated than that shown in Fig. 4, the auxiliary springs C'_1 and C'_2 being applied separately, instead of simultaneously, depending upon load distribution. Variation of the effective length of the axle arms as the suspension deflects gives a certain amount of non-linearity of the load-deflection characteristic. The layout was modified in 1955. In the new arrangement, the main springs operate in tension instead of in compression, while the auxiliary springs have been replaced by rubber buffers.

Fig. 5 shows the equalizing suspension used by Packard. The torsion bar springs 1, extending along each side of the frame, are common to the front and the rear wheels. A conventional front suspension layout has been adopted. At the rear, the tractive effort is transmitted by two trailing arms 2, which are linked to the main torsion bars 1, as well as to the short torsion bars 3. Without the short bars, the pitching stiffness of the suspension would be zero and the system unstable. The ends of the torsion bars 3 are connected by levers to a servo-motor 4, which maintains the body constantly in a horizontal position, irrespective of the load

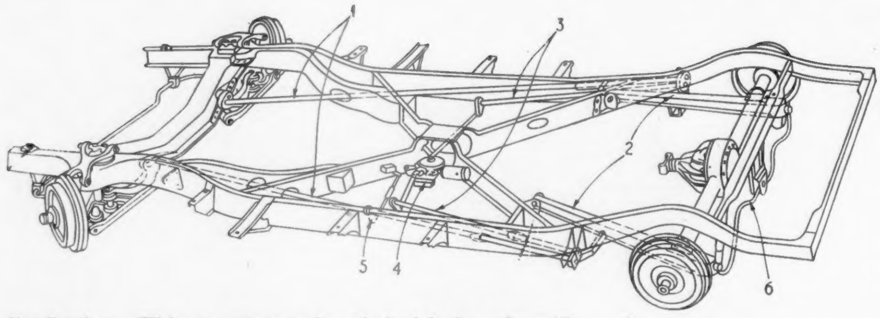
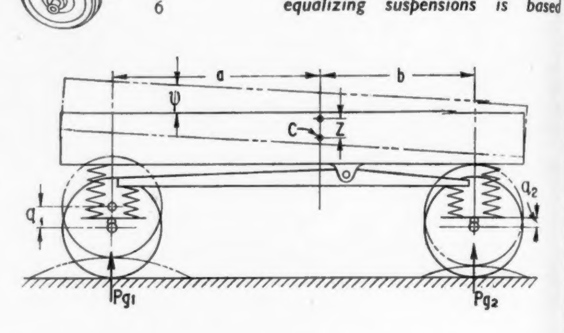


Fig. 5. Equalizing suspension system as used on the Packard cars. Without the short torsion bars 3, the pitching stiffness of the system would be zero and the car unstable

Fig. 6. Diagrammatic illustration on which the vibration theory of equalizing suspensions is based



distribution. This servo-motor is switched in by a lever 5, coupled to the main torsion bar. So long as the front and rear wheels are equally deflected, the position of the lever is unchanged. If the rear suspension is deflected more than that at the front the lever 5 is turned to the right, or if less, to the left. This brings the servo-motor into operation to rotate the torsion bars 3, thus raising or lowering the rear end of the body. To avoid operation due to transient deflections, the control system incorporates a relay, giving a time lag of 6 sec. The suspension incorporates a stabilizer 6, at the rear, while another anti-roll stabilizer is fitted at the front.

To determine the relative stiffness of suspensions of the Packard type, it is necessary to know the lever ratio between the wheels and the torsion bars, that is, the angular displacement of the wheel relative to the corresponding angular displacement of the torsion bar, assuming that the other wheel coupled to the same torsion bar remains unaffected. If the lever ratio between the front wheel and the main torsion bar is i_1 and between the rear wheel and the bar i_2, while the lever ratio between the rear wheel and auxiliary torsion bar is i_δ, then, as shown in Fig. 5, $i_2 = i_{2\delta}$, that is, both torsion bars are turned through the same angle.

Assuming $f_1 = 0$, while the rear wheels are displaced by f_2, the angle imparted to the torsion bars is $\alpha = i_2 f_2$; the moments are $M = C_\alpha \alpha$ and $M_\delta = C_{\alpha\delta}\alpha$, where C_α and $C_{\alpha\delta}$ are the stiffnesses of the torsion bars.

The reaction at the rear wheels is:

$$P_2 = (M + M_\delta)\, i_2$$
$$= (C_\alpha + C_{\alpha\delta})\alpha\, i_2$$
$$= (C_\alpha + C_{\alpha\delta})\, i_2^2 f_2$$

The reaction at the front wheels is:

$$P_1 = M i_1$$
$$= C_\alpha \alpha\, i_1$$
$$= C_\alpha\, i_1 i_2 f_2$$

and with regard to equation (1):

$$C_{22} = (C_\alpha + C_{\alpha\delta})\, i_2^2$$

and $\quad C_{12} = C_\alpha\, i_1 i_2$

Similarly, assuming $f_2 = 0$ and the displacement of the front wheels $= f_1$:

$$C_{11} = C_\alpha\, i_1^2$$

and $\quad C_{21} = C_\alpha\, i_1 i_2 = C_{12}$

Vibrations of a car with equalizing suspension

For the derivation of vibration data, the position of the sprung mass is defined by the co-ordinates z and ψ, Fig. 6, that is, the vertical displacement of the centre of gravity C and the angular displacement of the body in the vertical plane. The vertical displacement of the front and rear wheels, due to road irregularities, is q_1 and q_2 respectively. Effects of vibration dampers, unsprung masses and tyres can be allowed for in the equations but these effects have not been taken into consideration here, lest they lead to confusion. The derivation of these particular equations is based on the general equations (1). Because of this, the results apply to all types of equalizing suspensions besides the scheme incorporating a balance beam, as shown in Fig. 6.

To determine the vibrations of the car body, the following two equations must be used:

$$\left.\begin{aligned} M\ddot{z} - P_{g1} - P_{g2} &= 0 \\ I_0\ddot{\psi} - P_{g1}a + P_{g2}b &= 0 \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots (3)$$

where M is the mass of the body, $I_0 = M\rho^2$ moment of inertia of the body relative to the lateral axis through the centre of gravity, ρ the radius of gyration, a and b the distances from the centre of gravity to the front and rear axles, respectively, and P_{g1} and P_{g2} the dynamic load.

According to equation (1):

$$P_{g1} = C_{11} f_{g1} + C_{12} f_{g2}$$
$$P_{g2} = C_{22} f_{g2} + C_{21} f_{g1}$$

where f_{g1} and f_{g2} are the deflections of the front and rear springs under the dynamic load. Since:

$$f_{g1} = q_1 - z - a\psi$$
$$f_{g2} = q_2 - z + b\psi$$

Equation (3) furnishes:

$$M\ddot{z} + (C_{11} + C_{22} + C_{12} + C_{21})z + [a(C_{11} + C_{21}) - b(C_{22} + C_{12})]\psi = q_1(C_{11} + C_{21}) + q_2(C_{22} + C_{12})$$
$$I_0\ddot{\psi} + [a^2C_{11} + b^2C_{22} - ab(C_{12} + C_{21})]\psi + [a(C_{11} + C_{12}) - b(C_{22} + C_{21})]z = q_1(aC_{11} - bC_{21}) - q_2(bC_{22} - aC_{12}) \quad \ldots\ldots (4)$$

The left-hand sides of these equations refer to the free vibrations of the system, while the right-hand sides are determined by the road surface irregularities and can be regarded as a function of time. To consider the free vibrations of the system, equation (4) is used without its right-hand portion:

$$\ddot{z} + \omega_z^2 z + \eta_{c1}\psi = 0$$
$$\ddot{\psi} + \omega_\alpha^2\psi + \eta_{c2}z = 0 \quad \ldots\ldots\ldots\ldots (5)$$

where

$$\omega_z^2 = \frac{C_{11} + C_{22} + C_{12} + C_{21}}{M} \quad \ldots\ldots\ldots\ldots (6)$$

$$\omega_\alpha^2 = \frac{a^2C_{11} + b^2C_{22} - ab(C_{12} + C_{21})}{M\rho^2} \quad \ldots\ldots\ldots (7)$$

$$\eta_{c1} = \frac{a(C_{11} + C_{21}) - b(C_{22} + C_{12})}{M} \quad \ldots\ldots\ldots (8)$$

$$\eta_{c2} = \frac{a(C_{11} + C_{12}) - b(C_{22} + C_{21})}{M\rho^2} \quad \ldots\ldots\ldots (9)$$

The magnitudes ω_z and ω_α are the partial frequencies of the system, that is, the frequencies which would apply if one of the co-ordinates z or ψ is constant while the other is variable. If the magnitudes η_{c1} and η_{c2}, which are the coupling factors, are zero then the vertical and angular

oscillations do not affect each other. The equations (5) have the following solution:

$$\left.\begin{aligned}\psi &= B_1 \sin(\Omega_1 t+\alpha_1)+B_2 \sin(\Omega_2 t+\alpha_2)\\ z &= B_1\frac{\Omega_1-\omega_a^2}{\eta_{c2}}\sin(\Omega_1 t+\alpha_1)+B_2\frac{\Omega_2^2-\omega_a^2}{\eta_{c2}}\sin(\Omega_2 t+\alpha_2)\end{aligned}\right\} \quad (10)$$

where B_1, B_2, α_1 and α_2 are constants, and Ω_1 and Ω_2 are the natural frequencies of the system. The values of Ω_1 and Ω_2 are given by:

$$\Omega^2=\frac{1}{2}\left(\omega_z^2+\omega_a^2\pm\sqrt{(\omega_a^2-\omega_z^2)^2+4\eta_{c1}\eta_{c2}}\right) \quad \ldots\ldots\ldots(11)$$

Equations (5), (10), and (11) apply to conventional suspensions as well, while the magnitudes of ω_z, ω_a, η_{c1} and η_{c2} are different for equalizing suspensions. If the values of connecting stiffnesses C_{12} and C_{21} are equal to zero, then equations (4) to (11) are the same as for conventional layouts. As shown by equations (6) and (7), the presence of an equalizer of stiffness C_{12} and C_{21} between front and rear increases the frequency ω_z of vertical vibration and reduces that of pitching, ω_a. The reduction of the pitching frequency is of particular importance.

As shown by equation (7), it is possible to reduce the pitching frequency to any required value. If $a^2C_{11}+b^2C_{22}=ab(C_{12}+C_{21})$, the frequency can be zero, and this will be signified by instability owing to the absence of stiffness of the suspension, so far as pitching is concerned.

First example—car A:

Total sprung weight $G=2{,}000$ kg
Mass $M=G/g=2{\cdot}04$ kg-sec²/cm
Distance of centre of gravity from front axle $=a=0{\cdot}55L$ and from rear axle $=b=0{\cdot}45L$, where L is the wheelbase
Radius of gyration $\rho=0{\cdot}45L$
Suspension stiffness: front $=C_{11}=50$ kg/cm
rear $=C_{22}=50$ kg/cm
Stiffness of equalizing system $C_{12}=C_{21}=25$ kg/cm
According to equations (6) and (7);

$\omega_z^2=73{\cdot}5$ 1/sec²
and $\omega_a^2=31{\cdot}1$ 1/sec²
so that: $\omega_z=8{\cdot}55$ 1/sec
$=81{\cdot}6$ c/min
$\omega_a=5{\cdot}57$ 1/sec
$=53{\cdot}2$ c/min

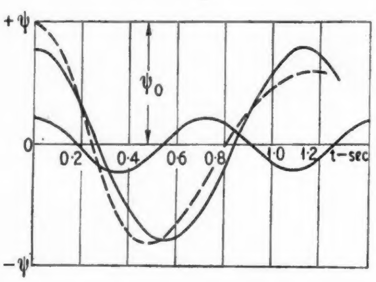
Furthermore, according to equations (11) and equations (8) and (9):

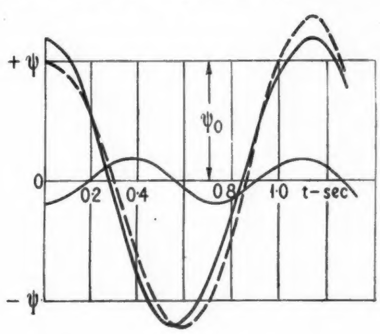
$\Omega_1^2=29{\cdot}7$ 1/sec²
and $\Omega_2^2=75{\cdot}0$ 1/sec²
so that $\Omega_1=5{\cdot}45$ 1/sec
$=52$ c/min
$\Omega_2=8{\cdot}64$ 1/sec
$=82{\cdot}5$ c/min

Thus, the natural frequencies Ω_1 and Ω_2 are nearly the same as the partial frequencies ω_a and ω_z, the value Ω_1 of the pitching frequency being particularly low.

Fig. 7. Left: The frequencies Ω_1 and Ω_2 are represented by the full lines, while the resultant vibration is shown by the dotted lines

Fig. 8. Right: Mode of vibration when the excitation is effected by vertical displacement of the suspension at the rear while that at the front is kept stationary





The actual natural frequencies Ω_1 and Ω_2 differ from the partial frequencies ω_z and ω_a, but for normal conditions as encountered with passenger cars these differences are negligible. As the values of the coupling factors η_{c1} and η_{c2} are increased the differences also increase. In turn, the magnitudes of the coupling factors increase as the difference between the static deflections of the front and rear suspensions becomes greater. Assuming that the static deflections are identical, that is, $f_{01}=f_{02}$, according to equation (2):

$$G_1(C_{22}+C_{21})=G_2(C_{11}+C_{12})$$

or, $$b(C_{22}+C_{21})=a(C_{11}+C_{12}) \quad \ldots\ldots\ldots\ldots\ldots\ldots (12)$$

Then, from equations (8) and (9), $\eta_{c1}=\eta_{c2}=0$

The frequencies Ω_1 and Ω_2 are identical with the partial frequencies, and the pitching of the body is independent of the vertical oscillations, see equation (5). Thus, the conditions for the independence of vertical and angular oscillations for the equalized suspension are the same as those for a conventional system.

Reduction of pitching frequency is the main advantage of equalized suspension. Pitching is particularly unpleasant to passengers. By lowering this frequency and thus reducing the angular acceleration, it is possible to improve riding qualities. On the other hand, the closer the exciting frequency is to the natural frequency of the vehicle suspension the greater is the amplitude of oscillation. The use of equalized suspensions enables the natural frequency to be reduced to a value at which resonance is not normally encountered. In these circumstances the vibration due to road irregularities is purely vertical. The following typical examples demonstrate the action of equalizing suspensions.

To prove that pitching characteristics are in the main determined by the frequency Ω, the pattern of vibrations is considered as follows. It is assumed that the vibrations are excited by an initial displacement of the vehicle body, that is $t=0$, $\psi=\psi_0$, $z=z_0$, $\dot\psi_0=0$ and $\dot z_0=0$. From the standard theory of vibration, it can be shown that in equation (10):

$$B_1=\frac{\psi_0(\Omega_2^2-\omega_a^2)-z_0\eta_{c2}}{\Omega_2^2-\Omega_1^2}$$

$$B_2=\frac{z_0\eta_{c2}-\psi_0(\Omega_1^2-\omega_a^2)}{\Omega_2^2-\Omega_1^2}$$

$$\alpha_1=\alpha_2=\pi/2$$

If, for example, only the front suspension is displaced and the rear one remains unaffected, then the vertical displacement of the centre of gravity z_0, and the angular displacement of the body ψ_0 can be obtained from the following equation:

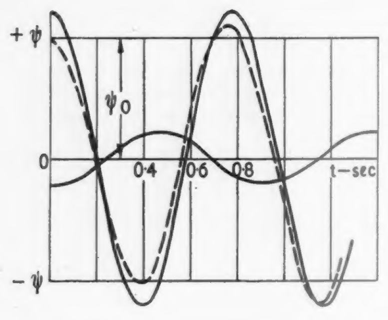
$$z_0=\psi_0 b$$

For the car A of the above example, $B_1=0{\cdot}788\psi_0$ and $B_2=0{\cdot}212\psi_0$; that is, the pitching, in accordance with equation (10) is:

$$\psi=\psi_0\,(0{\cdot}788\cos 5{\cdot}45t+0{\cdot}212\cos 8{\cdot}64t)$$

This expression shows that the frequency of pitching is low; the amplitude of the lower frequency Ω_1 is about four times higher than the amplitude of the higher frequency Ω_2, Fig. 7. The frequencies Ω_1 and Ω_2 are shown by full lines in Fig. 7 while the resultant vibration is shown dotted.

If the vibrations are excited by displacement of the rear suspension while the front is kept stationary, Fig. 8, then, it



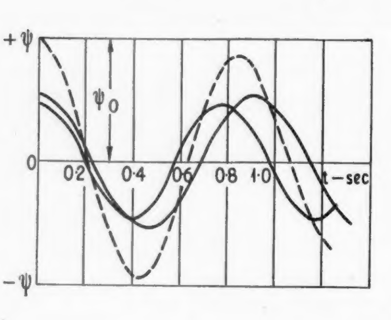


Fig. 9. Left: Pitching vibrations excited by displacement of the front suspension only, car B

Fig. 10. Right: Pitching vibrations excited by displacement of the rear suspension only, car B

can be shown, from the well-known theory, that

$$z_0 = -\psi_0 a$$

and, consequently:

$$B_1 = 1{\cdot}188\psi_0$$

$$\text{and } B_2 = -0{\cdot}188\psi_0$$

$$\psi = \psi_0\,(1{\cdot}188 \ \cos\ 5{\cdot}45t - 0{\cdot}188\ \cos\ 8{\cdot}64t)$$

Here, too, the oscillations are of a low frequency and in practice both are damped by the friction in the system. It can be shown that the results are similar if the vibrations are excited by a velocity $\dot{\psi}_0$ and $\dot{z}_0$, as encountered by rapid passage of road irregularities, instead of by an initial displacement ψ_0 and $\dot{z}_0$.

Second example—car B.

It is assumed that car B is similar to car A, but the magnitudes C_{12} and C_{21} are zero, that is, the suspension is of the normal type. Then the stiffness of the front suspension of car B is C_{11} and that of the rear is C_{22}. The following data apply to car B:

Total sprung weight $G = 2{,}000$ kg
Mass $M = G/g = 2{\cdot}04$ kg-sec²/cm
Distance from centre of gravity from front axle
$= a = 0{\cdot}55L$
and from rear axle $= b = 0{\cdot}45L$
Radius of gyration $\rho = 0{\cdot}45L$
Spring stiffness: front $C_n = 50$ kg/cm
rear $C_3 = 50$ kg/cm

The partial frequencies are:

$$\omega_z^2 = \frac{C_n + C_3}{M}$$
$$= 49{\cdot}0 \ 1/\text{sec}^2$$
$$\omega_z = 7{\cdot}0 \ 1/\text{sec}$$
$$= 66{\cdot}9 \ \text{c/min}$$
$$\omega_a^2 = \frac{a^2C_n + b^2C_3}{M\rho^2}$$
$$= 61{\cdot}0 \ 1/\text{sec}^2$$
$$\omega_a = 7{\cdot}8 \ 1/\text{sec}$$
$$= 74{\cdot}5 \ \text{c/min}$$

The coupling coefficients are:

$$\eta_{c1} = \frac{aC_n - bC_3}{M}$$
$$= 2{\cdot}45L$$
$$\eta_{c2} = \frac{aC_n - bC_3}{M\rho^2}$$
$$= 12{\cdot}1 \ 1/L$$

and the natural frequencies of body oscillations:

$$\Omega_1^2 = \tfrac{1}{2}\left(\omega_z^2 + \omega_a^2 - \sqrt{(\omega_a^2 - \omega_z^2)^2 + 4\eta_{c1}\eta_{c2}}\right)$$
$$= 46{\cdot}9 \ 1/\text{sec}^2$$
$$\Omega_1 = 6{\cdot}85 \ 1/\text{sec}$$
$$= 65{\cdot}3 \ \text{c/min}$$
$$\Omega_2^2 = \tfrac{1}{2}\left(\omega_z^2 + \omega_a^2 + \sqrt{(\omega_a^2 - \omega_z^2)^2 + 4\eta_{c1}\eta_{c2}}\right)$$
$$= 65{\cdot}2 \ 1/\text{sec}^2$$
$$\Omega_2 = 8{\cdot}06 \ 1/\text{sec}$$
$$= 77 \ \text{c/min}$$

Thus, with the equalizing suspension, the value of Ω_2 is slightly increased whilst Ω_1 is substantially reduced. As with the first example, it is possible to determine the pattern of pitching vibrations excited by various conditions. If the oscillations are excited by displacement of the front suspension, Fig. 9, then:

$$\psi = \psi_0\,(1{\cdot}206\ \cos\ 8{\cdot}06t - 0{\cdot}206\ \cos\ 6{\cdot}85t)$$

On the other hand, if they are due to displacement of the rear suspension, Fig. 10, then:

$$\psi = \psi_0\,(0{\cdot}458\ \cos\ 8{\cdot}06t + 0{\cdot}542\ \cos\ 6{\cdot}85t)$$

The resultant expressions and curves show that pitching oscillations of car B take place at a higher frequency Ω_2, and that the frequencies Ω_2 and Ω_1 are similar. In all cases, the resultant frequencies are higher than for car A.

With the vehicle incorporating equalized suspension, a low pitching frequency is obtained together with a relatively small static deflection of the suspension. This is an important feature. To illustrate this point, the static deflections f_{01} and f_{02} of the front and rear suspensions can be determined for cars A and B. From equation (2): for car A, $f_{01} = 9{\cdot}3$ cm and $f_{02} = 17{\cdot}3$ cm, while for car B, $f_{01} = 18{\cdot}0$ cm and $f_{02} = 22{\cdot}0$ cm.

By altering the relationship between the stiffnesses C_{11} and C_{22} it is possible to reduce still further the static deflection f_{02} of the rear wheels of car A without increasing the frequency of pitching. With equalized suspension, it is not possible to determine, from the static deflections alone, the natural frequencies or riding qualities of the vehicle.

In the examples previously discussed, it was assumed that the suspension stiffness values were the same for both the conventional and the equalized suspension, that is, $C_n = C_{11}$, and $C_3 = C_{22}$. Also, the frequency of partial pitching oscillations ω_a of the equalized system was much lower and the partial frequency of bouncing ω_z was slightly higher than with the conventional arrangement. It is possible, however, with an equalized suspension, for the mean static deflection to be the same as with a corresponding conventional suspension. This condition can be obtained by reducing the stiffness values, $C_{11} < C_n$, and $C_{22} < C_3$, of the equalized system, so that the frequency ω_z will not be increased and ω_a will be reduced even more. A further step in this direction has been taken on the latest Packard cars. In these, the natural frequency of bouncing is only 54 c/min, while that of pitching is 40 c/min.

Assume that the forced oscillations of a car with equalized suspension are induced by passage over a sinusoidal obstacle, that is, in equation (4) $q_1 = q_0 \ \sin\ pt$, where q_0 is the height of the obstacle and p the frequency of excitation. Then:

$$z + \omega_z^2 z + \eta_{c1}\psi = \frac{q_0\,(C_{11} + C_{21})}{M}\ \sin pt \ldots\ldots\ldots\ldots\ (13)$$

$$\ddot{\psi} + \omega a^2\psi + \eta_{c2} z = \frac{q_0\,(aC_{11} - bC_{21})}{M\rho^2}\sin pt \ldots\ldots\ldots\ (14)$$

The right-hand sides of equations (13) and (14) represent the excitation forces. Equation (14) shows that the excitation moment of pitching oscillation, for a given obstacle height q_0, can be reduced at the expense of the difference $(aC_{11} - bC_{21})$. By choosing a sufficiently high value for the coupling stiffness

C_{21}, it is possible to reduce the excitation moment as much as is desirable. The solution of equations (13) and (14) is:

$$z = z_m \sin pt$$

and

$$\psi = \psi_m \sin pt$$

Differentiating and introducing these expressions in equations (13) and (14):

$$z_m(\omega_z^2 - p^2) + \psi_m \eta_{c1} = \frac{q_0(C_{11}+C_{21})}{M}$$

$$z_m \eta_{c2} + \psi_m(\omega_a^2 - p^2) = \frac{q_0(aC_{11}-bC_{21})}{M\rho^2}$$

and from this:

$$\psi_m = \frac{\dfrac{q_0(C_{11}+C_{21})}{M}\eta_{c2} - \dfrac{q_0(aC_{11}-bC_{21})}{M\rho^2}(\omega_z^2-p^2)}{\eta_{c1}\eta_{c2} - (\omega_a^2-p^2)(\omega_z^2-p^2)}$$

If the static deflections are equal at the front and the rear, $\eta_{c1} = \eta_{c2} = 0$, as shown previously, and the amplitude of forced vibration is:

$$\psi_m = \frac{q_0(aC_{11}-bC_{21})}{M\rho^2\left(1-\dfrac{p^2}{\omega_a^2}\right)\omega_a^2}$$

Introducing the expression for ωa^2 from equations (7), and (12):

$$\psi_m = \frac{q_0}{1-\dfrac{p^2}{\omega_a^2}} \cdot \frac{aC_{11}-bC_{21}}{a(aC_{11}-bC_{21})+b(bC_{22}-aC_{21})}$$

$$= \frac{q_0}{\left(1-\dfrac{p^2}{\omega_a^2}\right)} L \qquad (15)$$

Equation (15) shows that for a given forced frequency p and natural frequency ω_a, the amplitude of pitching oscillations ψ_m will be independent of whether the suspension arrangement is of the conventional or equalized type. The effectiveness of the equalizing suspension is due to its ability to reduce the frequency ω_a. Although equation (15) is based on the condition of equal static deflections at the front and rear, the fundamental considerations are the same for other front:rear deflection ratios.

Braking or accelerating

Alteration of the static deflection owing to the application of braking or acceleration loads causes the body to assume a new position, both with regard to height and angle. Assuming that the load on the front wheels is increased by ΔG_1, then in accordance with equation (2), the static deflections will be increased by the following amount:

$$\left.\begin{aligned} \Delta f_{01} &= \frac{C_{22}\Delta G_1}{C_{11}C_{22}-C_{12}C_{21}} \\ \Delta f_{02} &= -\frac{C_{21}\Delta G_1}{C_{11}C_{22}-C_{12}C_{21}} \end{aligned}\right\} \qquad (16)$$

The minus sign indicates that the deflection of the rear suspension is reduced as the load on the front wheels is increased. To calculate the additional angular displacement, the following expression is used:

$$\Delta\psi = \frac{\Delta f_{02} - \Delta f_{01}}{L}$$

$$= -\frac{C_{22}+C_{21}}{C_{11}C_{22}-C_{12}C_{21}} \frac{\Delta G_1}{L} \qquad (17)$$

An increase ΔG_2 of the load on the rear wheels results in:

$$\left.\begin{aligned} \Delta f_{01} &= -\frac{C_{12}\Delta G_2}{C_{11}C_{22}-C_{12}C_{21}} \\ \Delta f_{02} &= \frac{C_{11}\Delta G_2}{C_{11}C_{22}-C_{12}C_{21}} \end{aligned}\right\} \qquad (16a)$$

$$\Delta\psi = \frac{\Delta f_{02} - \Delta f_{01}}{L}$$

$$= \frac{C_{11}+C_{12}}{C_{11}C_{22}-C_{12}C_{21}} \cdot \frac{\Delta G_2}{L} \qquad (17a)$$

Equations (16) and (17) show that the additional deflections, Δf_{01} and Δf_{02}, and the angular displacement $\Delta\psi$ depend to a considerable extent on the stiffness of the equalizing system C_{12} and C_{21}; increasing the stiffness results in an increase in Δf_{01} and Δf_{02} and a substantial increase of the angle $\Delta\psi$. With a conventional suspension system in which $C_{12} = C_{21} = 0$, the additional deflection and angular displacement of the body due to a change in the front:rear weight distribution is less than with an equalized suspension arrangement.

The above statements can be substantiated by the following example. Assume that both car A and car B have an additional load $\Delta G_2 = 150$ kg on the rear suspension. From equations (16a) and 17a):

For car A: $C_{11} = C_{22} = 50$ kg/cm
$C_{12} = C_{21} = 25$ kg/cm
$L = 250$ cm

Therefore: $\Delta f_{01} = -2{\cdot}0$ cm
$\Delta f_{02} = 4{\cdot}0$ cm
$\Delta\psi = 0{\cdot}024$
$= 1$ deg 20 min

For car B: $C_{11} = C_{22} = C_n = C_3 = 50$ kg/cm
$C_{12} = C_{21} = 0$
$L = 250$ cm

Therefore: $\Delta f_{01} = 0$ cm
$\Delta f_{02} = 3{\cdot}0$ cm
$\Delta\psi = 0{\cdot}012$
$= 0$ deg 40 min

Thus, with equalized suspension, the angle of tilt is twice that with a conventional system.

The sensitivity of the equalized suspension to the load distribution between the front and rear wheels constitutes its main disadvantage and limits the possibility of increasing the stiffness of the couplings C_{12} and C_{21}. If these stiffness values are of a similar order to the stiffnesses C_{11} and C_{22}, of the suspension, then even a slight alteration of the load distribution between the front and rear wheels leads to a substantial deflection of the suspension and tilting of the body. For a light car in the laden condition, the weight on the rear wheels is substantially greater than in the empty condition, and if it has an equalized suspension, this load tends to cause the rear end to drop substantially.

These considerations show that the automatic adjusting device used on the 1955 Packard cars to maintain the body in the horizontal position was introduced to compensate for the shortcomings of the equalizing suspension. The Citroën 2CV does not incorporate such a device. In the unladen condition, the body of the vehicle is tilted well forward; when loaded, it assumes an approximately horizontal attitude.

When a car is braked, its front wheels are of course, subjected to an additional vertical load, while the loading on the rear wheels is relieved. For a constant brake effort T at the wheels, the change in the vertical reactions at the front and rear wheels can be expressed approximately as follows:

$$\Delta P_1 = \frac{Th}{L}$$

$$\Delta P_2 = -\frac{Th}{L}$$

where h is the height of the centre of gravity of the vehicle and L its wheelbase. With an equalized suspension, the angle of tilt and the resultant redistribution of vertical reactions due to braking can be calculated from equations (17) and (17a):

$$\Delta\psi_T = -\frac{C_{22}+C_{21}}{C_{11}C_{22}-C_{12}C_{21}}\frac{\Delta P_1}{L} + \frac{C_{11}+C_{12}}{C_{11}C_{22}-C_{12}C_{21}}\frac{\Delta P_2}{L}$$

$$= -\frac{C_{11}+C_{22}+C_{12}+C_{21}}{C_{11}C_{22}-C_{12}C_{21}}\frac{Th}{L^2} \qquad (18)$$

For example, if a brake force $T = 0{\cdot}6G = 1{,}200$ kg, is applied to car A, in which $h = 70$ cm, $L = 250$ cm, then $\Delta\psi_T = 0{\cdot}1075 = 6$ deg 10 min. With car B, incorporating a conventional suspension, and having the same values of T, h

and L, the value of $\Delta\psi_T = 0{\cdot}0536 = 3$ deg 5 min. As was the case with the uneven load distribution, the body tilt due to braking is considerably greater with the equalized suspension. Similar results are obtained with forces due to acceleration.

Equation (18) gives the angular position of the body when the brakes are applied and the vertical reactions at the wheels are changed. With some layouts brake torque also has a substantial effect on the angle of tilt. To reduce excessive forward tilting of the body when the brakes are applied, suspension layouts that tend to tilt the sprung mass backwards under brake torque should be used. With the Citroën 2CV, Fig. 2, the rear wheels are carried on trailing arms. Brake torque, therefore, tends to lower the rear end of the body. The front wheels are carried by forward-extending arms so the brake torque tends to lift the front end of the body. Thus, the brake torque reaction helps to resist the nose-down tendency when the car is braked. With the Packard, Fig. 5, trailing arms locate the rear wheels, so in this case also the brake torque tends to lower the rear end of the body and opposes forward tilt of the sprung mass.

Conclusion

1. Equalized suspension can substantially improve the riding qualities of cars, mainly because of the reduction of the natural frequency of pitching. It leads to a reduction of the angular acceleration due to pitching and tends to prevent "galloping" in resonance with road irregularities.

2. The natural frequencies of an equalizing suspension system can be estimated by the method outlined.

3. The main disadvantage of equalizing suspensions is that they give only a small resistance to changes in attitude due to redistribution of the load, braking or acceleration.

Automatic compensating devices can be used to reduce the tilting of the body. However, this adds to the complication of the system. Reduction of body tilting during acceleration or braking can be effected by the adoption of suspension layouts that tend to react against this undesirable tendency; the layout adopted on the Citroën 2CV is a noteworthy example.

Recent Publications

Brief Reviews of Current Technical Books

Body Engineering

By Sidney F. Page, M.I.B.C.A.M., M.M.I.A., A.M.I.E.D.
London: CHAPMAN AND HALL LTD., 37, Essex Street, W.C.2. 1956. $8\frac{3}{4} \times 5\frac{1}{2}$. 190 pp. Price 25s.

The second edition of this book, which has now been published, is divided into eight sections. After an introduction dealing with the motor car body and construction trends, there are sections on the materials used in bodywork, and on bodywork design and drafting. Chapters on sheet-metal projection, caravans and trailers, bodywork interior, and practical bodywork construction are also included. The book ends with a constructional supplement, comprising a series of tables and brief descriptions of new constructional methods. This supplement occupies five pages. The book should be of use in particular to apprentices interested in body design and layout.

Maintenance of High Speed Diesel Engines

By A. W. Judge, A.R.C.S., D.I.C., Wh.Sc., A.M.I.Mech.E., A.M.I.A.E.
London: CHAPMAN AND HALL LTD., 37, Essex Street, W.C.2. 1956. $8\frac{3}{4} \times 5\frac{1}{2}$. 422 pp. Price 56s.

The fourth edition of this book has just been issued. It consists of much the same material as the third edition, which has been largely rewritten and a considerable number of new illustrations introduced, to bring the information up-to-date. Also, the chapters have been re-arranged to some extent and two new ones added. One of the new sections is on engine testing after overhaul.

This work is, of course, primarily for diesel engineers, service echanics, operators, apprentices, and others concerned with the practical aspects of the compression ignition engine. Although the treatment of the subject is mostly concerned with the automobile type of compression ignition engine, much of the information given is also applicable to others, for example, stationary, marine, agricultural, locomotive and tractor engines.

The chapters are headed: Working principles of compression ignition engines; The compression ignition engine in practice; General maintenance of compression ignition engines; The engine components—cylinders; The pistons and piston rings; The valves and valve seatings; The connecting rod and crankshaft; The fuel-injection system; Fuel-injection nozzles and nozzle holders; Timing the fuel pump; Starting procedure and running troubles; Heater plugs, filters, air cleaners, etc.; Engine testing after overhaul. There is an index at the end of the book.

Practical Solution of Torsional Vibration Problems, Vol. I.

By W. Ker Wilson, D.Sc., Ph.D., M.Sc., Wh.Ex., M.I.Mech.E.
London: CHAPMAN AND HALL LTD., 37, Essex Street, W.C.2. 1956. $8\frac{3}{4} \times 5\frac{1}{2}$. 704 pp. Price 105s.

Neither Dr. Ker Wilson nor his book, of which this is the third edition, need introduction; the first edition was published in 1935 and quickly became recognized as a standard work of reference on this subject. Since the first publication of this work, the pace of engineering development has quickened to such an extent that vibration study can no longer be regarded as a wholly specialized activity. A thorough knowledge at least of the fundamentals is essential for automobile engineers.

In this new edition, the text has been completely rewritten to bring the subject matter up-to-date, but the original aim, which was at providing an adequately comprehensive treatment in a form suitable for everyday reference, has been retained. There are still two volumes in the complete work, but the text has been re-arranged so that each volume is now self-contained, in that it deals with a particular aspect of the subject. In this first volume are included frequency calculations and analyses of the characteristics of different types of oscillating systems. The book also incorporates considerable additions, particularly in the sections dealing with geared systems and systems containing distributed masses. Natural frequency determination by the effective inertia method is discussed in greater detail, and the application of this method to cases of coupled vibration are treated more comprehensively.

The chapters are headed as follows: Simple systems; Frequency tabulations; Multi-mass systems; Simple geared systems; More complicated geared systems; Coupling effects in geared systems; Effective inertia method—concentrated mass systems; Effective inertia method—distributed mass systems; Coupled torsional and flexural vibration in engine systems; Equivalent masses; Equivalent shafts; and Official requirements relating to vibration. At the end of the book there is an extensive bibliography occupying ten pages. This is followed by a list of tables and a list of numerical examples. The work is concluded with name and subject indexes. Listed in the name index are the manufacturers, and scientific, research and other organizations, as well as the authors mentioned both in the book and in its bibliography.

Simple but adequate treatments of the mathematical problems have been given, and there is an abundance of fully worked numerical examples from current engineering applications. In short, the book is unusually comprehensive, and treats the subject in a way that will appeal to practising engineers as well as to students.

Lubrication of Bearings

By F. T. Barwell, Ph.D., B.Sc.(Eng.), Wh.Sc., M.I.Mech.E., M.I.Loco.E., A.M.I.E.E.
London: BUTTERWORTHS SCIENTIFIC PUBLICATIONS LTD., 88, Kingsway, W.C.2. 1956. $8\frac{3}{4} \times 5\frac{1}{2}$. 292 pp. Price 50s.

This is a subject that has been dealt with in many research reports, both theoretical and experimental; nevertheless, little progress, so far as appreciation of the principles underlying the behaviour of bearings is concerned, has been made since Reynolds first propounded the basic theory of hydrodynamic lubrication, which was towards the end of the nineteenth century. This is, perhaps, indicative of the complexity of the subject, and points to the need for further research.

In the book under review, the Author has assembled a selection of the most significant of the researches that have been published during the last fifty years, and the form of presentation is such

that it is suitable for both research workers and practising engineers. The Author has set himself two aims: one has been at the provision of data in such a form as to be suitable for direct application to the design of bearings, and the second has been at outlining the work done so far, and so to form a basis from which others can embark on further research.

A procedure for the design of bearings is described, which has not appeared elsewhere, and which enables proportions of journal bearings to be determined by the application of simple formulae. Apart from considering the common types of bearings, the Author has discussed at some length the more complicated lubrication conditions occurring in engines and between gear teeth. Fundamental considerations related to friction and wear are also reviewed in the light of their influence on the selection of lubricants and bearing materials.

The chapters are headed as follows: Surfaces; Modes of lubrication; Wear; Viscosity; Principles of hydrodynamic lubrication; Units and non-dimensional presentation; Experimental work on journal bearings; Oil flow in journal bearings; Thermal considerations; Hydrodynamic design of partial bearings; Tentative approach to bearing design; Journal bearings subject to variable loading—oscillatory behaviour; Thrust bearings; Concentrated contact bearings; Externally pressurized bearings; Reciprocating sliders and oscillating bearings; Materials for bearings; Lubricants for particular applications; Arrangements for testing bearings; and Conclusion. There are two appendices, and name and subject indexes.

Abacs or Nomograms

By A. Giet.

London: ILIFFE AND SONS LTD., Dorset House, Stamford Street, S.E.1. 1956. $8\frac{3}{4}\times5\frac{1}{2}$. 235 pp. Price 35s.

Most engineers have made use of nomograms at some time in their careers and are well aware of the fact that charts of this type are very convenient for application when a formula has to be solved repeatedly for several sets of variables. However, even among those who habitually employ nomograms, only a small proportion know how to construct them for their own use. Most of the literature on this subject hitherto published has been written for mathematicians and is difficult for the practical engineer to comprehend, but the subject has been treated in a simple manner in the new work, which will therefore be welcomed by engineers as well as by those more skilled in the use of advanced mathematics.

This book, which is a translation from French by H. D. Phippen, M.A., B.Sc. and is edited by J. W. Head, M.A., is essentially practical. It not only demonstrates the many and varied applications of the abac or nomogram, but also shows how these charts can be constructed even by those without highly specialized mathematical knowledge. The book deals with both Cartesian abacs and alignment charts, and contains a large number of practical examples from mechanical and electrical engineering and physics. Few formulae are outside the scope of nomograms, if skilfully chosen auxiliary variables are introduced.

The work is divided into five main sections. Of these, the first deals with relations between two variables; it contains two chapters, one headed Diagrams and the other Scales. The second section is headed Cartesian abacs, and contains four chapters headed: Relations involving three variables; Superimposed Cartesian abacs for three variables; Superimposed Cartesian abacs for four variables; Superimposed Cartesian abacs for relations involving n variables ($n>4$). There are six chapters in the third section, which covers alignment charts. These chapters are: Parallel co-ordinates, Graphical representation of a relation of the first degree between three variables; Standard forms of alignment chart for relations between three variables; Charts for four variables; Charts for three variables requiring the use of auxiliary variables; Combined charts for several equations involving common variables. The fourth section is on alignment charts not based on parallel co-ordinates. It is subdivided into chapters entitled: Charts with concurrent scales; Circular abacs; and Double-alignment charts for a relation between four variables of the form $f_1f_2=f_3f_4$. The last section deals with relations between n variables ($n>4$). In this section, there are only two chapters, one on relations of the form $f_1+f_2+f_3+\ldots\ldots+f_n=O$, and the other on relations of the form $f_1 . f_2 . f_3 \ldots\ldots f_{n-1}=f_n$. Finally there is an appendix on the choice of methods and choice of abac. This is a most useful work of reference for engineers engaged on design, research and production.

Theory and Practice of Lubrication for Engineers

By Dudley D. Fuller.

London: CHAPMAN AND HALL LTD., 37, Essex Street, W.C.2. 1956. $9\frac{1}{2}\times6$. 432 pp. Price 84s.

In this work, the aim has been at giving a balanced treatment of the theory and practice of lubrication; the basic concepts are arranged in a manner which the author, from his teaching experience, has found to be most practical. He begins by laying a sound foundation in the theory of lubrication. Then the basic principles are applied to a wide range of lubrication problems. Many numerical examples are given to illustrate the discussions of the general problems. Among the large number of types of bearings discussed are those for machine tools, turbines, generators, large motors, rolling mills, ball mills and centrifuges.

Direct references are made to outstanding examples of recent research in lubrication, and much of the material included should stimulate further research work. Research techniques previously employed and the results obtained are fully discussed. The work is useful both as a text book for an organized course and for individual study.

The hydrodynamic and hydrostatic theories have been developed from readily understood principles. In this work the simplest mathematical methods have been employed in preference to those which may be more elegant but which would appeal to a smaller number of readers. Emphasis has been placed on the restrictive assumptions upon which the hydrodynamic and hydrostatic theories are based. This is because these assumptions, in many instances, strictly limit the fields of application of these theories.

In this book, the concept of hydrodynamic lubrication has been generalized and applied to many different forms, rather than limited to that of the geometric wedge as found in tilting pad and plain journal bearings. Extensive coverage is given to hydrostatic lubrication since this aspect of the science has almost numberless and still unexplored applications, ranging from air bearings in the lightest of instruments to heavy rolling mill bearings.

The chapters are headed: Fundamentals of viscosity and flow; Viscosity and its variables; Hydrostatic lubrication; Hydrostatic squeeze films; Hydrodynamic lubrication; Hydrodynamic lubrication of journal bearings; Friction and power losses in journal bearings; Some typical industrial bearings; Air-Lubricated bearings; Dry friction; Boundary friction; and Bearing materials.

Automobile Engine Testing and Tuning

By S. G. Mundy, M.I.Mech.E., M.I.E.E., M.I.M.I.

London: GEORGE NEWNES LTD., Tower House, Southampton Street, W.C.2. 1956. $9\frac{1}{2}\times6$. 264 pp. Price 35s.

This book is written mainly for those who, for one reason or another, are unable to acquire a better understanding of the modern automobile engine and its electrical auxiliary equipment from text books that give full theoretical treatment. In other words, it deals with the principles of operation of the automobile engine in a practical way and with a minimum of theory. The aim has been at providing a course in the testing and tuning of the modern automobile engine. Instruction is given in the recognition and correction of engine performance faults and electrical defects.

The chapters are headed: Modern automobile service; The modern automobile engine; Fundamental principles of the internal combustion engine; Engine vacuum; Engine compression; The carburation system; Servicing the carburettor division; The A.B.C. of ignition; The ignition circuit; Sparking plugs: The charging circuit—generators and voltage regulators; Battery service; The principle of scientific engine-diagnosis; Equipment for scientific engine-diagnosis; Systematic engine-diagnosis procedure; and The application of scientific engine-diagnosis. From this outline of the ground covered, it can be seen that the work is of the type that would appeal to mechanics and to private car owners.

Automotive Mechanics

By William H. Crouse.

London: MCGRAW-HILL PUBLISHING CO. LTD., 95, Farringdon Street, E.C.4. 1956. $9\frac{1}{2}\times6$. 726 pp. Price 39s.

This, the third edition of "Automotive Mechanics," provides a complete course, covering the theory of operation, construction, maintenance, repair, dismantling, and adjustment of automotive components. With few exceptions, it deals with all the subjects of the courses listed in the Automotive Industry—Vocational Education Conference's "Standards for Automotive Service Instruction in Schools." The exceptions include features such as welding and bodywork, which are special subjects that require separate treatment.

One purpose of revising the book was to include in it new developments such as the latest V-eight engines, valve rotators, sodium-cooled valves, American 12-volt electrical equipment, four-barrel carburettors, LPG fuel systems, automatic transmissions, power-assisted steering and braking, air conditioning, and tubeless tyres. Another aim has been at improvement of the clarity of the work. It includes much new material on fundamentals such as physical principles, engine measurements, chemistry of combustion, hydraulics, and front suspension geometry. Comprehensive sections and charts dealing with trouble analysis are included. The book should be particularly useful to those who are concerned with the maintenance of American automobiles and trucks, but is also of general interest to all who wish to know more about the mechanical features of these vehicles.

Hypoid Bevel Double-Drive Bogie

Kirkstall 24-ton Heavy-Duty Assembly, with Concentric Reduction Gearing in the Wheel Hubs; A Matching Driven Front Axle is Available for Off-the-Road 6×6 Vehicles

SOME four or five years ago, Kirkstall Forge Engineering Ltd., of Leeds, introduced a series of hypoid-bevel rear axles for medium and heavy vehicles. Subsequently, they have developed this type of axle for double-drive twin-axle bogies designed to carry weights up to 53,700 lb (24 tons). These assemblies are now being supplied in increasing numbers. Since the name Kirkstall is so intimately associated with heavy-duty axles of the worm-drive type, the reasons forming the background to the development of this range of hypoid axles are of interest.

Worm-drive is a peculiarly British feature of heavy vehicle design and for many years has been regarded almost as the standard layout on both goods and the larger passenger vehicles. Some features of this type of design are open to criticism, but by and large, it gives excellent results. It is suitable for the transmission of heavy torques, is silent in operation and adaptable with regard to both layout and the range of ratios that can be provided within a common dimensional structure. However, it is also true that with this type of gear, the designer is not faced too quickly by limitations due to excessive tooth loading as the ratio is lowered.

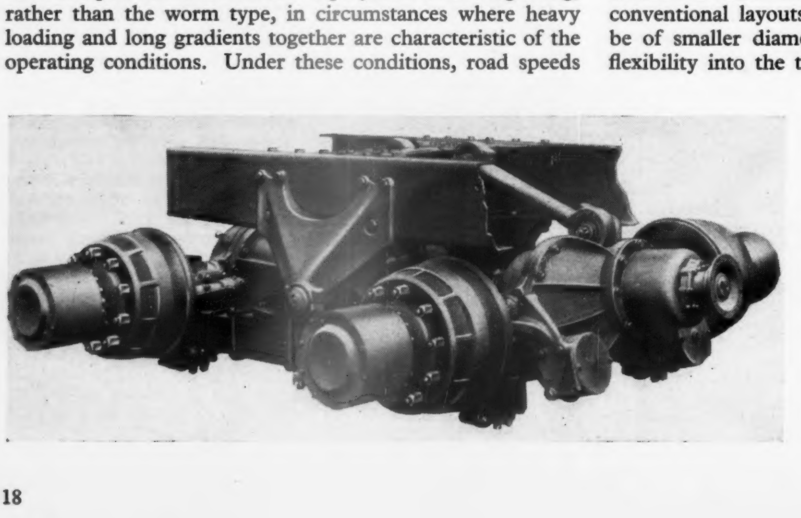
In recent years, vehicles with worm-driven axles have been exported in considerable numbers. Many have gone to countries where American influence has tended to predominate and where maintenance staffs are either American or have been trained on American vehicles; this influence is particularly marked among the major companies engaged in oil well and pipe line operations. To such staffs, the worm-drive axle is an unfamiliar construction. It was therefore considered that for markets where the maintenance men, because of their greater familiarity with bevel-driven axles, are likely to be more favourably disposed towards this type, the availability of such equipment would enhance the export appeal of the heavy-duty vehicles being produced by British manufacturers.

Development of the hypoid-drive arrangement, therefore, does not indicate any diminution of faith in the worm-drive type of axle. Rather is it indicative of the policy of Kirkstall Forge Engineering Ltd., which is to meet the requirements of different markets, as against adhering to a single type in face of strong regional preferences. Technically, of course, there is a point in favour of the employment of bevel gearing, rather than the worm type, in circumstances where heavy loading and long gradients together are characteristic of the operating conditions. Under these conditions, road speeds are generally low and the efficiency of a worm-drive is also low by comparison with that of a bevel drive; this consideration may add force to a preference, which otherwise appears to be based on personal prejudice.

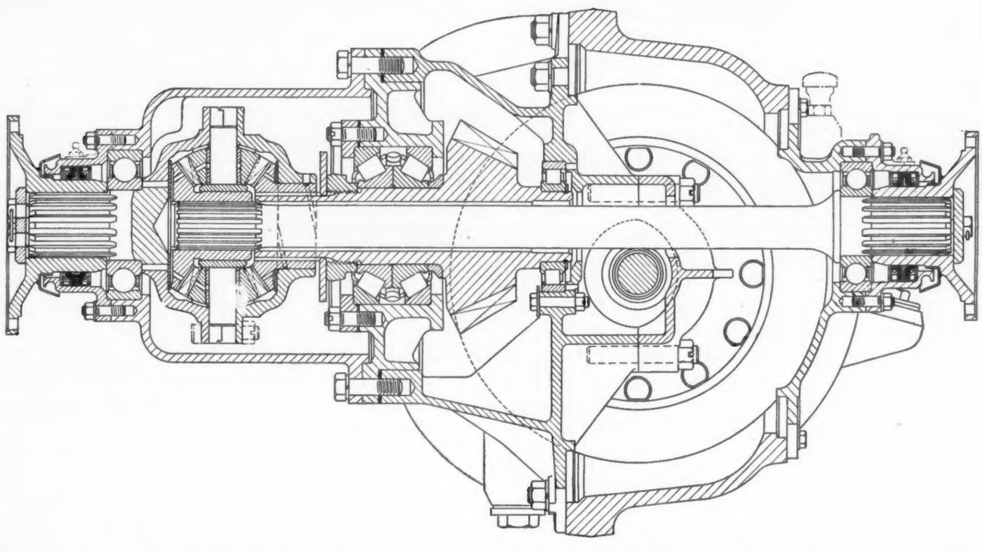
Since most heavy-duty, large capacity vehicles exported from this country are of the six-wheel type with double-driven bogies, the worm-drive arrangement could not be discarded lightly. The mounting of the worm above or below the worm wheel makes it particularly easy to couple the two worm shafts, either with or without a third differential. Here again, there are two schools of thought, but it can hardly be denied that, in terms of reduced tyre wear alone, the third differential pays handsome dividends. From the practical aspect, because of its simplicity, the worm-drive layout is attractive for use with a third differential. Generally, the mechanism of the third differential is carried on the nose of the worm housing of the leading axle, and the worm shaft of that axle is hollow so that a drive shaft can be passed through it. The short cardan shaft between the two axles is coupled to a flange on the end of this drive shaft.

With a plain bevel drive arrangement, obviously the bevel shaft of the leading axle cannot be extended rearwardly, to couple it to the rear axle of a double-drive bogie, since it must intersect the axle shaft axis. Therefore, either some form of transfer drive is needed to bridge the axles or a suitable arrangement of twin propeller shafts must be used. These are not particularly attractive solutions to the problem. The hypoid bevel appears to offer greater scope, because the offset of the pinion shaft relative to the half shafts makes a through-drive practicable. However, the dimensional latitude is, of course, strictly limited.

Having arrived at these conclusions, Kirkstall decided to use the hypoid layout in conjunction with another of their earlier design features to form the arrangement that they have now patented. An advantage of this design is that a large reduction ratio in the hypoid drive is unnecessary, so the bevel pinion can be of appreciable size; therefore, the pinion can be bored to clear a drive shaft that is passed through it. This shaft carries the coupling for the connection to the rearmost axle. Another advantage is that the small initial reduction at the bevel, which is made possible by the subsequent multiplication at the hubs, gives a torque loading on the axle shaft that is lower than that obtained with more conventional layouts. This means that the half shafts can be of smaller diameter to introduce a degree of torsional flexibility into the transmission line to each of the wheels.



Kirkstall double-drive double-reduction heavy-duty hypoid bevel bogie assembly



From the third differential, the drive to the rear axle passes through the hollow spindle of the hypoid pinion and thence over the half shaft

The final reduction, by which the hypoid ratio is multiplied, is provided by a concentric spur gear system engaging with an internally-toothed ring in the wheel hub. This is not strictly an epicyclic gear, although it has equivalent elements. On the end of the half shaft is a sun pinion, with which are meshed a ring gear, secured to the hub, and three planet-like spur gears. These three spur gears rotate on roller bearings on studs mounted in a carrier fixed to the outer end of the axle tube.

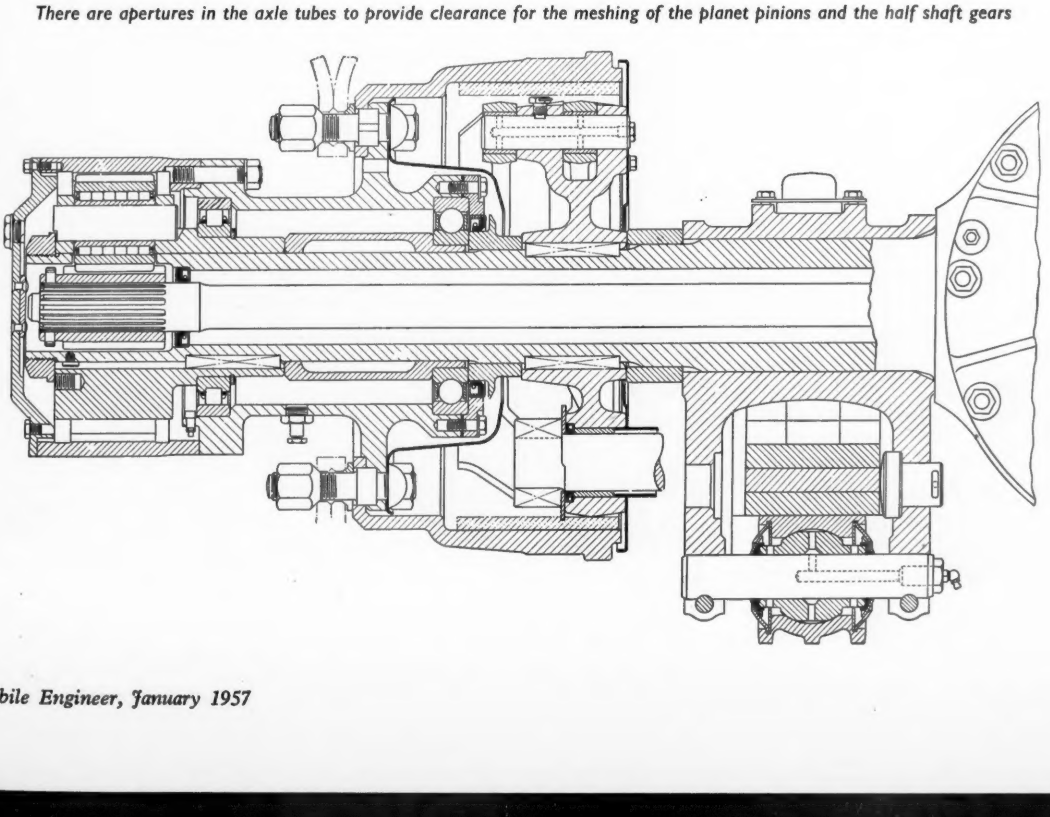
In the hub, the reduction ratio is 3·5 : 1, while overall ratios, between the limits of 6·59 : 1 and 10·18 : 1, can be provided by variation in the hypoid ratio from 1·88 : 1 to 2·90 : 1. The hypoid bevel has 11 teeth for the lowest ratio and 13 teeth for the most commonly used final drive ratio of 8·89 : 1. A pinion offset of $1\frac{11}{16}$ in has been adopted.

The axle casings are of the familiar, Kirkstall one-piece forged type with detachable front and rear covers. Housed in the front cover are the taper roller bearings that carry the hypoid bevel pinion. The third differential is in a separate housing mounted on the front cover. All the differentials are of the four-star bevel type with spherical backed pinions.

For assembly purposes, the hole through the hypoid bevel spindle is big enough to allow the forward end of the extension shaft, which is $1\frac{3}{4}$ in diameter over its 16 splines, to be passed through it. The hypoid bevel pinion shaft is carried by two taper roller bearings, supplemented by a parallel type roller bearing at the rear. Since the hub gearing reverses the direction of rotation of the axle shafts, the hypoid drive is set above the axle, instead of below. An advantage of the above-centre pinion arrangement is that it enables a particularly favourable propeller shaft line to be adopted, especially in types of vehicle for which this bogie has been primarily designed.

A widely spaced pair of taper roller bearings carry the

There are apertures in the axle tubes to provide clearance for the meshing of the planet pinions and the half shaft gears



differential cage of the axle. Both the inner and outer ends of the axle shafts are $1\frac{3}{4}$ in diameter over the splines. The wheel hub is located by a large single-row ball bearing at its inner end, and there is a parallel type roller bearing at its outer end. In this application, taper roller bearings are not desirable because, if they are used, special measures must be taken to ensure concentricity with the axle tube under running conditions, otherwise the hub gears may be noisy and other troubles are liable to be experienced. A noteworthy feature of the arrangement is that the gear housings neither obstruct wheel changing nor do they lead to any difficulty in the withdrawal of the brake drums.

The driving connection from the torque-dividing differential in the nose of the leading axle casing is a shaft splined into the boss of the forward of the two differential gears. It passes through the hollow hypoid bevel of the leading axle, and thence clear above the axle shaft and through the rear cover of the forward axle. In this cover is a housing for a single row ball bearing retained by a cover plate bolted on to the rear of the housing. The back-to-back oil seal assembly, round the boss of the coupling flange, is housed in the cover plate.

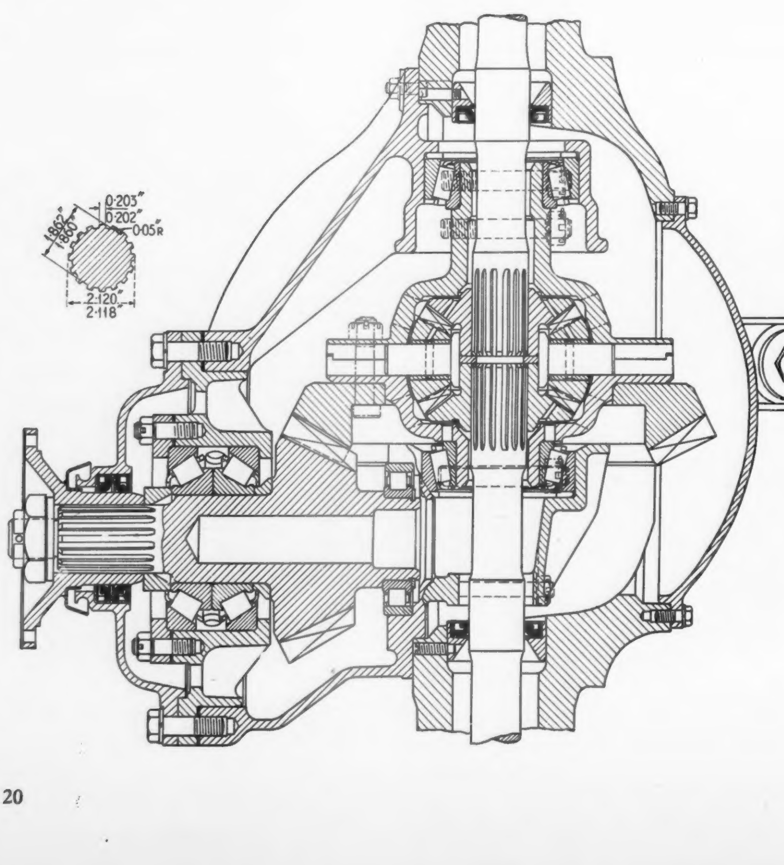
A noteworthy feature of the assembly is the large number of components common to the front and rear axles of the bogie. Among these are the main forging, the front cover and the differential carrier. The detachable housing for the third differential, of course, is not fitted to the rear axle, and the shaft of the hypoid bevel pinion is therefore shorter.

As in the front axle, an opposed pair of taper roller bearings and a single-row plain roller bearing carry the hypoid bevel pinion. Although the rear end of the pinion shaft in the back axle is hollow, its forward splined end, carrying the cardan shaft coupling, is left solid. The rear cover of the second axle is of a simple pattern, since its sole function is to close the end of the unit. It can, of course, be removed to give access to the gears.

Well established Kirkstall practice is followed in the general design of the bogie, which is of the trunnion type. Inverted, reverse camber, semi-elliptic springs are employed. They are 5 in wide, and there is one spring assembly on each side of the chassis. A collar round the centre of the trunnion tube, provides for location. Each tube is gripped on top of the box-like spring-clamp. The spring assembly swings on needle roller bearings, the outer races of which are hardened steel cups flanged at their open ends and blind at their opposite ends. Hardened steel buttons are spigoted into the ends of the hollow trunnion to take the side loads between the trunnion and the inner faces of the blind ends of the bearing cups.

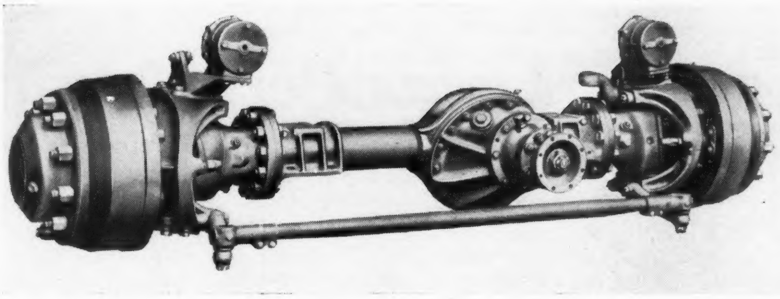
The trunnion and spring assembly on each side is retained in the bracket on the frame by engagement of the bearing cups in semi-circular seatings in the brackets, where they are clamped by caps. Whereas the trunnions are designed to help to give torsional rigidity to the bogie assembly, the spring ends accommodate considerable angular movement. To this end, the axle casing carries on each side a substantially proportioned bracket which incorporates a downward pointing fork below and a bump pad seating above. Bridging the lower ends of the fork is a pin which is passed through a $3\frac{1}{2}$ in diameter ball. This ball is free to slide on the pin, and to rotate in a divided block. It is case hardened and chromium plated. The top face of the upper half of the divided block is flat, while the lower half is a cap secured to the upper portion by studs and nuts. This assembly is free to oscillate within wide limits about the pin, but its side movement is limited by hardened steel buttons between it and the inner faces of the fork. This end assembly for the spring is the subject of a patent.

On each block, a pin is passed through lugs on the vertical face nearest the trunnion, to carry the eye of the master leaf of the spring. The second leaf is extended over the flat upper surface of the block and bent downwards. Drilled through the bent-down end is a hole to receive a projecting boss on the block. The vertical load of the vehicle is applied through the spring to the flat upper surface of the end blocks, the main leaf reacting principally the driving and brake



A noteworthy feature of the bogie assembly is the large number of components common to the front and rear axles. From this illustration of the rear axle, it can be seen that the mounting of the hypoid bevel pinion is the same as that in the front axle

This powered front axle is specially designed to match the capacity and characteristics of the bogie used at the rear



torques. Supplementary torque and radius arms are provided above the axles. They are a pair of rubber bushed rods connected at one end to levers integral with and above the axle casing, and at the other end to upward extensions of the frame trunnion brackets.

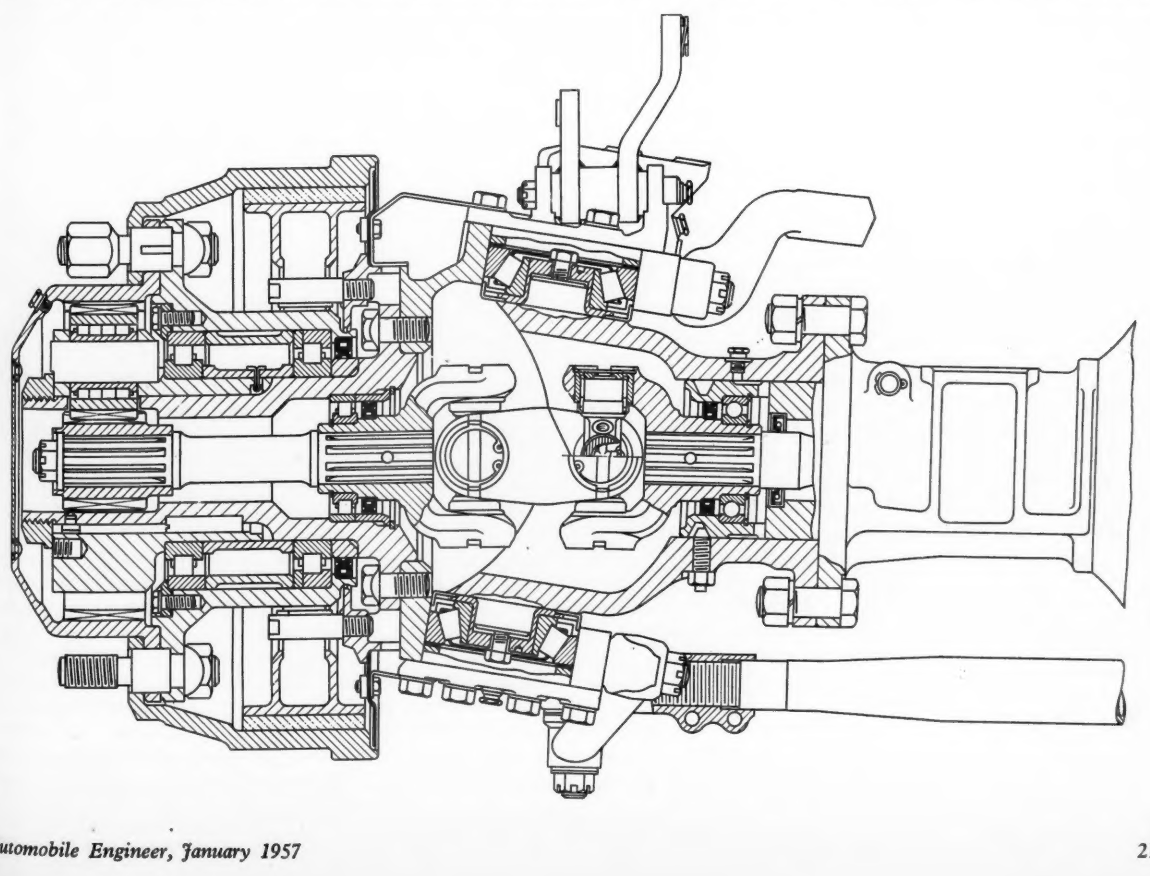
With 11.00—20 in tyres fitted, the overall width of the assembly is 8 ft and the load carrying capacity of the complete bogie is 53,700 lb (24 tons). The gross vehicle-weight, with a trailer, is 135,000 lb (60 tons). For certain special duties, larger tyres are fitted, but these are not for the purpose of increasing load capacity: they are 14.00—20 in and are for flotation; that is, operation on sand and other unstable surfaces. With such tyres the overall width becomes 9 ft 3 in, which, of course, is outside the legal limits for road operation in this country.

It is stated that the torque rating of the bogie is 60,000 lb-ft but that this figure is quoted subject to certain conditions, among which are that the distribution of output between the four wheels is reasonably uniform. The weight, less tyres, rims and propeller shaft, is 5,400 lb. In the front axle of the bogie, the oil capacity is 18 pt, while in the rear one it is 16 pt. Brakes of Kirkstall design are incorporated; they are $16\frac{1}{2}$ in diameter and the shoes are 6 in wide. Expansion is effected by a snail cam actuated by an air pressure system.

An interesting further development of the application of double reduction transmission gearing is a powered front axle, specially designed to match the capacity and characteristics of the bogie. In this axle, a normal spiral bevel gear is employed in conjunction with a hub gear arrangement of the same type as at the rear. The steering swivels, which are massive forgings, are articulated on large diameter taper roller bearings. Short members, which form the common inner links of two needle roller universal joints, transmit the drive from the axle shafts to the hubs. The universal joints accommodate the angular motion for steering, which is equivalent to a total of 60 deg included angle, although this, of course, is not quite equally divided between the two locks.

The hub reduction gears are similar to those on the bogie axles, but parallel roller bearings are used throughout. Larger brake drums are employed, their diameter being 17 in, but the shoe width is only 4 in. The load capacity of the front axle is 16,700 lb, which, of course, is 7·5 tons.

A short coupling, with needle-roller type universal joints at its ends, transmits the drive to each wheel hub of the front axle for use with the bogie



Lucas Petrol Injection

Brief Outline of the General Principles of the System

FULL details of the Lucas petrol injection system are not yet available, partly because it has so far been employed only in special applications. Therefore, the following description and the accompanying diagrammatic illustrations are intended to do no more than to outline the general principles of the arrangement, as applied to the Jaguar sports cars that were entered during the last season at Sebring, Nurburgring, Reims and le Mans. This system, of course, could be adapted for application to any multi-cylinder petrol engine.

The problem of petrol injection has been under investigation by Lucas engineers for the last fifteen years, initially for aircraft engines and later with the object of exploring the practicability of its successful application to automobile engines. In the course of these investigations much ground has been covered, and the system employed on the Jaguar sports cars already mentioned, is the outcome of a great deal of intensive work.

At an early stage in the investigations, it was found that the conventional type of multi-cylinder injection pump, as used for diesel engines, was in some instances unsuitable for employment on spark ignition engines burning petrol. With a spark ignition engine, the requirements, with regard to accurate control of the fuel:air ratio and its equal distribution to all the cylinders are difficult to meet with a pump of the conventional multi-cylinder type. Success is more easily attained if the system has a common metering unit serving each cylinder in turn.

Apart from the problems involved in the design of the pump itself, the exact manner of the introduction of the fuel into the cylinder has been the subject of intensive investigation. One of the main advantages of petrol injection is that it can give uniformity of mixture distribution between all the cylinders of a multi-cylinder engine. To attain this end, it is necessary, of course, to have individual injection, either into the cylinder itself or into the port close to it. Investigations showed that for the relatively small cylinders of automobile engines, injection into the actual cylinder head is not practicable: therefore, in the system that has been developed, the fuel is injected as an atomized spray into the ingoing air stream in or near the induction port. This not only ensures equality of distribution of the fuel, but also takes advantage of the rapid movement of the air stream to assist atomization.

Components of the fuel system

Basically, the fuel system comprises a supply line, pressurized at approximately 100 lb/in^2, feeding fuel through a metering unit to each cylinder in turn. This supply can be pressurized either by an engine-driven pump or by a separate electrically-driven pump, the latter being preferable although possibly slightly more expensive. The components of the Lucas six-cylinder petrol injection installation, as fitted to Jaguar cars, are shown schematically in Fig. 1. They comprise an electrically-driven high pressure fuel pump incorporating the main filter, a secondary filter, a metering distributor and six injector nozzles.

Petrol is drawn from an anti-surge well in the base of the tank, and is filtered before entering the pump. It is then passed under pressure through the secondary filter to the distributor. In this unit, the petrol is metered and passed in a timed sequence to each nozzle in turn. The function of the nozzles is to inject the fuel as a very fine spray into the ingoing air streams. A mixture control device, mounted on the metering distributor, is connected pneumatically with a balance pipe joining each induction tract. This device regulates the amount of petrol injected, so that the fuel delivery is proportional to induction pressure.

Fuel pump

The fuel supply pump is preferably fitted either in or near to the tank, so that its operation is unaffected by heat radiated from the engine. It consists of a simple, gear type pump, driven by a built-in, series-wound motor. A relief valve regulates the pressure to 100 lb/in^2, surplus fuel being returned to the tank. An advantage of this system is that the fuel in all the pipes, from the tank to the injection nozzles and engine, is pressurized as soon as the ignition switch is turned on. This effectively prevents any trouble due to vapour lock, since any pockets of vapour formed while the vehicle is standing are collapsed immediately. A sectional view of the pump is shown in Fig. 2, which shows the gear pump in the base of the unit.

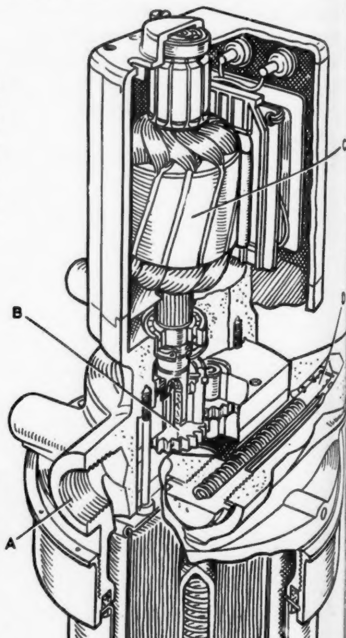
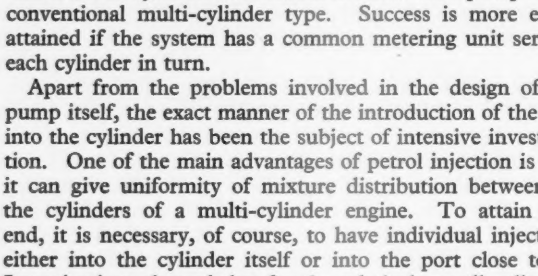


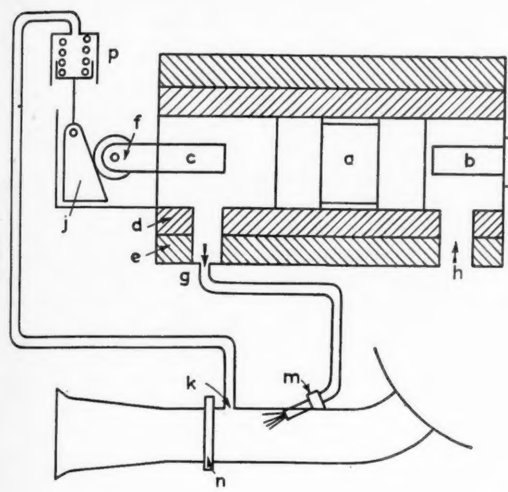
Fig. 2. Diagram showing the electric motor, pump and filter assembly, sectioned to show the gear type pump between the motor and the main fuel filter

A—Fuel inlet; B—Gear type pump; C—Electric motor; D—Pressure relief valve; E—Main fuel filter



A—Petrol tank; B—Spill return to tank; C—Electric motor; D—Pressure gauge; E—Datum adjuster; F—Vacuum tapping from inlet manifolds; G—Mixture control; H—Injector nozzles; J—Oil pump; K—Drive shaft, rotating at half engine speed; L—Metering distributor; M—Secondary filter; N—Fuel pump; P—Main filter

Fig. 1. Below: Components of the Lucas petrol injection installation for six-cylinder power units



a—Shuttle; b—Fixed stop; c—Mixture control stop; f—Follower; g—Fuel delivery to nozzle; h—Fuel inlet under pressure; j—Control wedge; k—Tapping to mixture control; m—Injector nozzle; p—Piston

Fig. 3. Illustration showing the principle of operation of the mixture control device

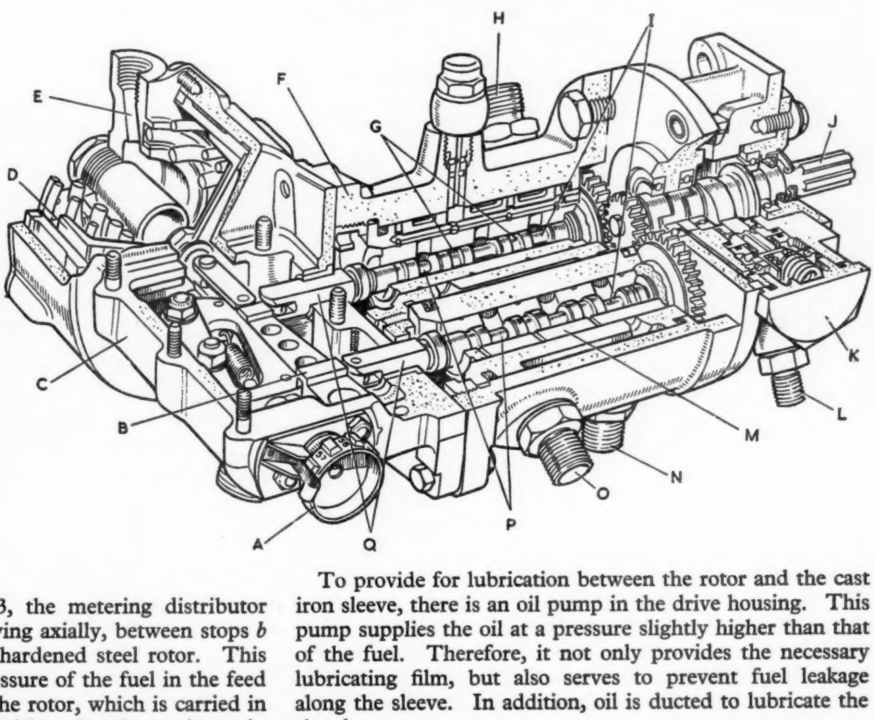


Fig. 4. Sketch showing a sectioned metering unit for the Jaguar six-cylinder engines

A—Datum adjuster; B—Control wedge; C—Mixture control; D—Piston; E—Vacuum tapping from inlet manifolds; F—Metering distributor; G—Shuttles; H—To number 1 cylinder; I—Fixed stops; J—Drive shaft; K—Oil pump; L—Engine oil inlet; M—Rotor; N—To number 6 cylinder; O—To number 5 cylinder; P—Control stops; Q—Mixture control followers

Metering distributor

As can be seen from Fig. 3, the metering distributor incorporates a small shuttle moving axially, between stops *b* and *c*, in the lapped bore of a hardened steel rotor. This movement is effected by the pressure of the fuel in the feed pipe from the supply pump. The rotor, which is carried in a stationary cast iron sleeve, is driven by the engine. At each end of the rotor and its housing-sleeve, there are radial ports drilled in such a way that, as the rotor revolves, first the port at one end is opened to the pressure feed line, while that at the other end is connected to the delivery line serving one of the cylinders; then as the rotor turns further, the pressure feed line is connected to what was previously the delivery end, while the opposite end is opened to the delivery line to another cylinder. Thus the chambers at each end of the shuttle are alternately filled and emptied as the shuttle moves to and fro between the two stops: at each movement, the displaced fuel is injected into the appropriate cylinder. The quantity of fuel injected is equal to the area of the bore, in which the shuttle moves, multiplied by the travel between the two stops; thus an identical quantity of fuel is injected into each of the cylinders in turn. Control of the quantity injected is effected by making one of these stops movable.

In order to vary the quantity of fuel injected per stroke and to keep the fuel:air ratio substantially constant over the working range, a mixture control device is incorporated; this is also shown diagrammatically in Fig. 3. It consists of a piston *p*, subject to the induction pressure in the manifold or at the inlet valve, working against a spring and regulating the position of the control wedge *j*, against which abuts a movable stop in the distributor. The wedge is profiled in a way such that the amount of fuel injected per stroke is proportional to the absolute pressure in the manifold; thus, a constant fuel:air ratio is maintained.

In the high-speed six-cylinder metering unit developed for the Jaguar engines, the principle of operation is the same, but the mechanism is more elaborate. This unit is shown sectioned in Fig. 4. It can be seen that a double-rotor unit is employed. Each rotor, which serves three cylinders, contains two shuttles and has three double inlet-ports and three single outlet-ports. The successive opening and closing of ports in this unit is effected by driving the rotors at one-quarter engine speed. A datum adjuster is incorporated for making trimming adjustment to the mixture ratio. As a further refinement, adjustment devices can be incorporated to correct for atmospheric pressure and temperature variations, although in many cases these are unnecessary.

To provide for lubrication between the rotor and the cast iron sleeve, there is an oil pump in the drive housing. This pump supplies the oil at a pressure slightly higher than that of the fuel. Therefore, it not only provides the necessary lubricating film, but also serves to prevent fuel leakage along the sleeve. In addition, oil is ducted to lubricate the shuttles.

The sequence of shuttle movements for one rotor of a double unit is shown in Fig. 5. Each injection corresponds to an induction stroke of the engine and consequently, the second rotor, not shown in this illustration, must provide injections phased 120 deg of crankshaft rotation, after each injection effected by the first rotor. Petrol enters the metering distributor by a single inlet union and thence through the twin ports in each rotor to the appropriate shuttle chamber. It is then displaced, by the shuttle movements, through each of the six single outlet ports in turn to the injectors.

Injector nozzles

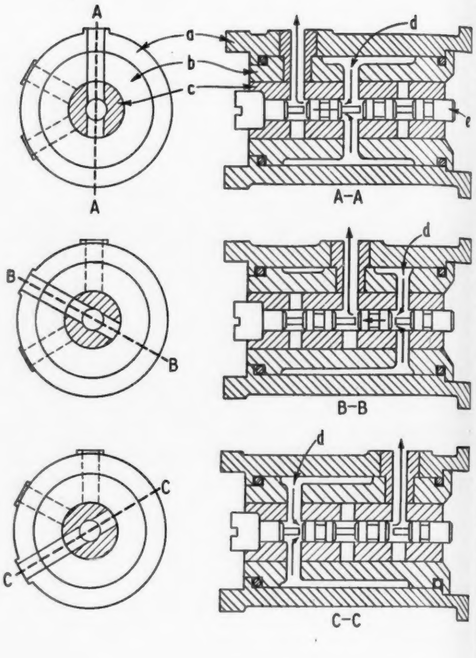
An injector nozzle is illustrated in Fig. 6. It comprises a spring-loaded poppet valve arranged to lift when the fuel pressure reaches a value of about 50 lb/in^2, and to spray atomized fuel upstream in each inlet port. Adequate mixing between the fuel droplets and the air stream is obtained by this arrangement, because the relative velocity between the fuel droplets and the air is high.

Inlet manifold and throttle

One of the chief advantages of a petrol injection system is that it makes possible the use of a manifold designed solely to carry a maximum flow of air to the cylinders. In the Lucas design of intake system used on Jaguar cars, the air intake is completely free of obstruction when the throttle is fully open; there is neither a choke tube nor a throttle spindle and plate assembly.

The throttle is a steel plate extending the full length of the cylinder block. This plate is carried on rollers, so that it can slide freely fore and aft, Fig. 6. It is actuated by a rack-and-pinion device coupled to the accelerator pedal. The plate has as many holes in it as there are engine cylinders. These holes are of the same diameter as the induction tracts and are so spaced that endwise movement of the plate regulates the size of the throttle apertures. By this method of air control, perfect throttle synchronization at all cylinders can be obtained.

Tuning to give a ram effect at certain speeds, is effected in this design by choice of both diameter and length of the intake trumpets. The performance characteristics of the engine can be altered at will, by fitting different trumpets, to make it suitable, for example, for a short twisty circuit where maximum acceleration through the gears is required, or a long fast circuit where maximum power has to be developed at high car-speeds.



a—Body; b—Sleeve; c—Rotor; d—Fuel entry; e—Fixed stop

Fig. 5. Sequence of shuttle movements for one rotor of a double-rotor metering unit

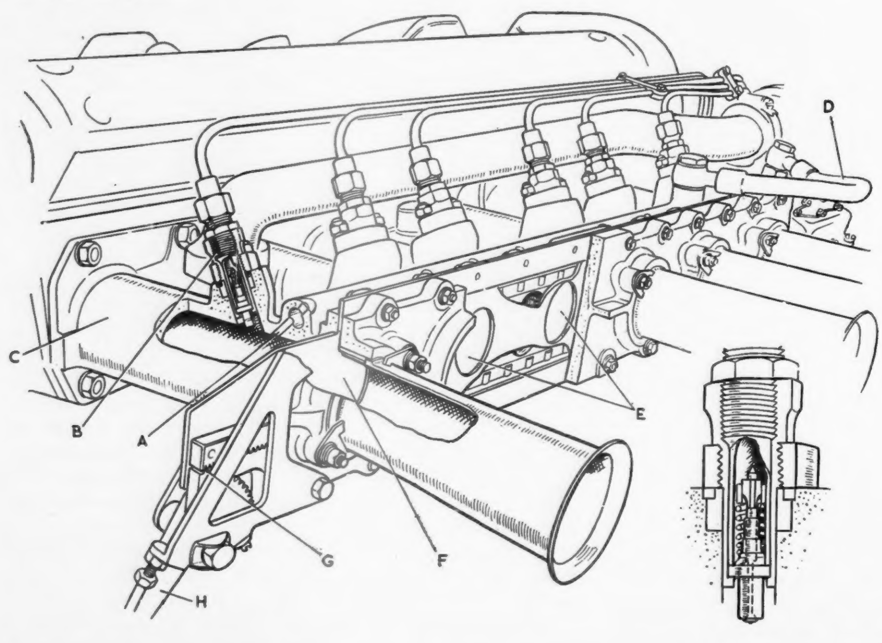


Fig. 6. In the installation used on the Jaguar engines, the throttle is a steel plate carried on rollers and extending the whole length of the cylinder block. A sectioned injector nozzle is shown to a larger scale in the scrap view on the right

A—Vacuum tapping; B—Injector nozzle; C—Induction tract; D—Vacuum tapping to mixture control; E—Air intake holes in slide; F—Throttle slide, shown in fully closed position; G—Rack and pinion; H—Connected to pedal

Possible future applications

While the system outlined has a fairly limited application on racing cars, it can readily be adapted to any multi-cylinder engine. The components that have been described are costly to manufacture, and such designs are never likely to be able to compete in cost with simple carburettors. If less complex versions of the injection equipment were to be used, however, there is reason to expect that the cost differential between the two would be considerably narrowed. Certainly, an eight-cylinder petrol injection installation of simple design might not compare unfavourably with the extremely complex four-barrel carburettors now being made for V-eight engines in the United States.

Because of the relatively low injection pressures and absence of cams and tappets, it would appear that the unit will not be noisy in operation. Lubrication and sealing problems have been solved effectively and simply. An important requirement, of course, is absolute cleanliness of the fuel, because the smallest particles of grit would tend to jam the shuttles; the filtration arrangements appear to be adequate to prevent this. The Lucas engineers are to be congratulated on both the originality of thought that has gone into the design of this unit and the relative simplicity of the installation.

As has previously been announced, a Licence and Engineering Assistance Agreement, covering petrol injection for automotive engines, has been made between Joseph Lucas (Industries) Ltd. and the Holley Carburetor Company of Detroit. The Agreement provides for a licence to the Holley Carburetor Company, under all Lucas fuel injection patents for automotive petrol engines, and the licensing of Lucas, under all Holley fuel injection patents outside the United States of America. Further clauses in the Agreement provide for complete exchange of engineering knowledge and data between the American and British companies.

New Plant and Tools

Recent Developments in Production Equipment

Unit head drill

THE unit spindle-head is a convenient and highly adaptable element for building up special-purpose machines without the expense of a completely new design. A newcomer in this field, Fig. 1, has been introduced by W. J. Meddings Ltd., 16, Berkeley Street, London, W.1, in the form of a development of automatic drill-head unit-equipment. Known as the type H.G.H. head, it embodies the Pacera-Maxam air-feed mechanism to permit infinitely variable feeds combined with a fast approach and slow break-through feed, if required. Having a capacity of $\frac{1}{2}$ in diameter, in mild steel, the units can be supplied with a chuck or with a No. 1 or No. 2 Morse taper spindle.

As many units as may be required, or can be accommodated, may be linked together to operate in a sequence such as might be needed in the functioning of a rotary indexing table or a transfer jig. The spindle drive is by a $\frac{1}{2}$ h.p. motor and has a drill-stroke of $2\frac{7}{8}$ in. A spindle speed gearbox is used and, in conjunction with five sets of pick-off gears and a selection of motor speed or gears, it is possible to obtain speeds between 80 and 6,000 r.p.m.

The units can be mounted horizontally, vertically, or inclined at an angle. Wherever other than a horizontal mounting is used, the sight-feed oil reservoir seen at the rear of the head should be brought to the highst point of the installation by the addition of a length of copper conduit.

Fig. 2. Sykes helical cutter sharpening attachment
W. E. Sykes Ltd.

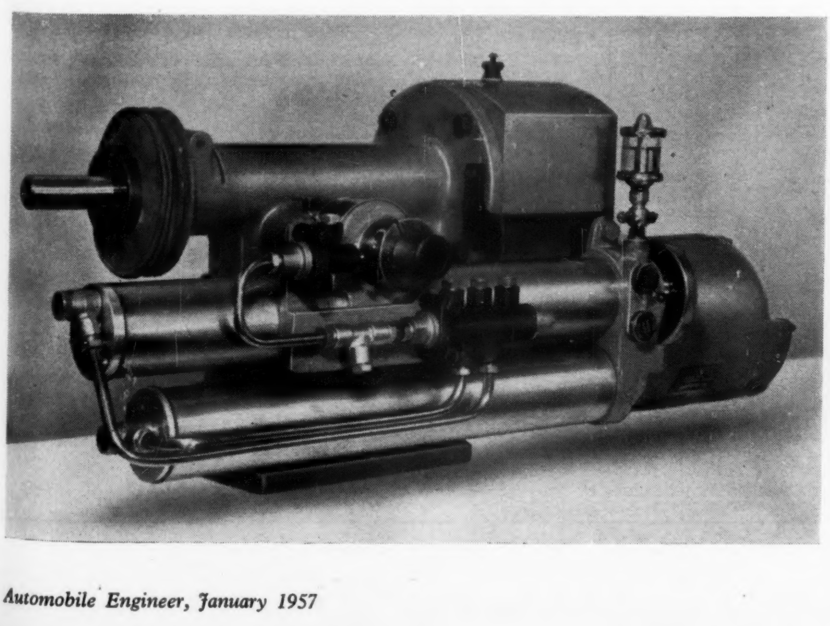
Helical cutter sharpening attachment

A NEW model of their helical cutter sharpening attachment for grinding involute cutting edges on gear shaper cutters, Fig. 2, is now produced by W. E. Sykes Ltd., of Staines, Middlesex. This is a development of the type SCA attachment and is suitable for cutters of the Sykes helical and double-helical pattern. The cutting edges of these cutters are in the same or parallel planes, a system which it is necessary to adopt when generating continuous-tooth double-helical gears. This practice is also recommended for precision gears having large helix angles and under conditions where there is a restriction on cutter run-out at the end of the operating stroke.

The operating instructions are simple and straightforward, and cutters can be sharpened with a near-mechanical routine on the part of one operator within a short space of time. Setting procedure consists of first clamping the attachment and wheel dresser unit on the machine table, and then adjusting a vernier scale, graduated in increments of 0·001 in, to give a swing-equivalent of half the base circle diameter, as marked on the reverse surface of the cutter. Following this initial setting of the attachment, the grinding wheel is dressed in accordance with detailed instructions supplied with the equipment. A plunger is used for locating the cutter between successive teeth and this is withdrawn for indexing by means of a lever. The actual regrinding of a tooth is done by swinging the indexing plunger sub-assembly radially between stop pins.

The manufacturers recommend grinding wheels of J to L bond and 36 to 46 grit for finish grinding operations, although this will depend, of course, on the condition of the machine.

Fig. 1. Pacera H.G.H. type unit drill head with infinitely variable feed
W. J. Meddings Ltd.



AXLE GEAR LUBRICANTS

Some Post-War Trends in the United States of America

ON the other side of the Atlantic, there were no major developments in gear lubrication from the end of the last war until 1952. Since then, there has been a great deal of experiment in this field, particularly on the problem of scoring. It is generally agreed that this type of failure occurs when metal is transferred from one gear to the other as localized welds are formed; high pressures and sliding velocities, accompanied by lubricant film failure, result in temperatures great enough to soften the metal and cause welding. Some of the work being carried out on this problem has been described recently, by C. M. Heinen of the Materials Laboratories Engineering Division of the Chrysler Corporation, at the National Fuels and Lubricants meeting of the Society of Automotive Engineers, held recently in Oklahoma.

At present, only rough approximations of the pressures and velocities that cause scoring can be made. In the absence of methods for calculation, the general approach has been to observe tests on axles operated in such a manner as to produce scoring.

It is well known that the heavier the load, the greater is the tendency for the oil film to be penetrated and, therefore, for potential welding surfaces to be brought together. Also, the greater the load, the larger the amount of work to be transmitted, and the greater the quantity of heat to be dissipated. High pressures have recently been produced by what are called bump tests, which have been used successfully to produce scoring. A feature common to most of these tests is the sudden application of an inertia load to the gears.

If scoring is considered as a welding phenomenon, it is apparent that if the surfaces are initially hot, only moderate pressures are required to cause scoring. A condition under which these high initial temperatures might be attained would be that experienced when the gears are running, even at relatively light loads, at speeds high enough to prevent the heat from having time to dissipate between revolutions. For a given output speed, this tendency would be the more marked the lower the axle ratio. This, of course, is because of higher pinion speeds.

A curious parallel between lorry and passenger car axle scoring at different road-speeds has been pointed out; pinion speed also appears to be a factor. Racing car experience, with high axle ratios, appears to indicate that for a given output speed, lower ratios result in less scoring. It has also been reported that scoring is prevalent under

Axle Lubrication Recommendations for Different Types of Car

Year	Factory fill	Drain plug	How drained	Type of oil	Temperature	Drain interval	Special Instructions
1952–6	A.S.	No	Cap screws	M.P.	All	None	Drain and refill not recommended—If necessary refer to dealer for special lubricant
1952–6	A.S.	No	Siphon	M.P.	Above 0 deg.F Below 0 deg.F	None	Drain and refill at overhaul only, except for low temperatures
1952–4	A.S.	No	Cap screws	M.P.	Above 0 deg.F Below 0 deg.F	None	Drain and refill for below 0 deg.F or severe service only
1955–6	A.S.	No	Siphon	M.P.	All	None	Drain and refill not recommended—SAE80 may be used for extended periods of extremely low temperatures
1952–4	A.S.	Yes	Plug	H.P. or M.P.	Above 10 deg.F Below 10 deg.F	10,000	None
1955–6	A.S.	Yes	Plug	M.P.	Above 10 deg.F Below 10 deg.F	None	Regular drain and refill not recommended—Special factory fill lubricant must be used for new gears during first 5,000 miles of operation
1952	A.S.	Some	Plug or Siphon	M.P.	All	10,000	None
1953–4–6	A.S.	Yes	Plug	M.P.	All	10,000	None
1955	A.S.	Yes	Plug	H.P.	All	10,000	None
1952–5	—	Yes	Plug	H.P.	All	None	Regular drain and refill not recommended—SAE80 may be used for consistently low temperatures
1952–6	M.P.	Yes	Plug	H.P.X.	All	10,000	None
1952–6	A.S.	Yes	Plug	M.P.	Above 10 deg.F	10,000	None
1952–6	A.S.	Yes	Plug	H.P.	All	10,000	None
1952	A.S.	No	Cap screw	M.P.	Above 10 deg.F Below 10 deg.F	None	Regular drain not recommended
1953–4	A.S.	No	Siphon	M.P.	Above 10 deg.F Below 10 deg.F	None	Regular drain not recommended
1955–6	—	No	Siphon	M.P.	All	None	Regular drain not recommended
1952–4	A.S.	Yes	Plug	H.P.	Above 10 deg.F Below 10 deg.F	Yearly	None
1955–6	A.S.	Yes	Plug	M.P.	Above 10 deg.F Below 10 deg.F	Yearly	None
1952–6	A.S.	No	Cap screws	H.P.	All	None	Regular drain and refill not recommended
1952–4	M.P.	Yes	Plug	H.P.	All	10,000	None
1955–6	M.P.	Yes	Plug	H.P.	Above 10 deg.F Below 10 deg.F	None	SAE80 not recommended for year-round use or for high-speed driving
1952–3	—	Yes	Plug	H.P.	Above 10 deg.F Below 10 deg.F	10,000	None
1954–6	—	Yes	Plug	H.P.	All	10,000	None
1952–6	M.P.	Yes	Plug	M.P.	Above 10 deg.F Below 10 deg.F	20,000	None

M.P.—*Multi-purpose gear lubricant* **H.P.**—*Hypoid gear lubricant* **H.P.X.**—*Hypoid gear lubricant (special)* **A.S.**—*Hypoid gear lubricant (active sulphur)*

operating conditions obtained on turnpike roads. Operational evidence in general seems to indicate that scoring is just as likely to occur as a result of a series of relatively light loads, as from a single heavy load: localized temperature build-up would explain this. High oil temperatures might lead to gear damage even though the mechanism were sound. It is interesting to see how recommendations for overcoming the problem have varied during the last four years.

Application problems

In the Table are given data on the axle lubrication of all American passenger cars during the last four years. The nature of the lubricants used explains some of the inconsistencies. Additive lubricants that give anti-weld coatings are required; but, unfortunately, they lead to other difficulties.

In a paper entitled "The Development of Automotive Gear Lubricants—Present and Future," presented to the A.S.L.E. at Milwaukee, in Nov., 1954, the authors T. P. Sands and J. L. Werner, state: "A proper degree of activity of these chemical additives is important. An inactive compound reacts neither quickly enough nor extensively enough to provide adequate protective reaction products. On the other hand, an over-active additive may attack the vital surface at relatively low temperatures, removing chemically combined metal directly, or forming reaction products that are too easily sheared. Either results in excessive wear."

Although there is some indication of considerable wear in the gears of passenger cars as a result of the use of high-activity lubricants, heavy loads that lead to high rates of wear are encountered most frequently in lorries. It has become apparent that universal or multi-purpose gear oils are not active enough chemically to form adequate films under peak torque conditions. Several solutions have been tried with a view to supplementing the film. They include:

1. Operation for a short period with active sulphur lubricant, followed by draining and filling with a multi-purpose gear oil.

2. Forming a phosphate film on the gears during production and using multi-purpose gear oil.

3. Using an active sulphur lubricant for the life of the vehicle.

It is the third recommendation that causes the most trouble from a marketing point of view. As has been indicated before, an active sulphur type of lubricant is not satisfactory for lorries, and since many of these vehicles are lubricated at service stations, two lubricants would have to be kept in stock. This increases costs and, what is even worse, leads to considerable confusion. Fear of this confusion at service stations has led seven manufacturers, representing about 70 per cent of the industry, to recommend that no change be made from the oil put in at the factory. These recommendations have been emphasized by the removal of the drain plug.

From the point of view of the lubricant manufacturer, this practice means a considerable loss of sales. In addition, there are disadvantages to the customer and the vehicle manufacturer, for it means that the gears are lubricated by oils contaminated with breakdown products, moisture, road abrasives, and metal particles. Moreover, the additives gradually become depleted.

There has been agreement that multi-purpose oils are needed. If oils currently available have been to some degree inadequate, it is logical to look for tests that must be adopted or developed to define a more suitable quality level.

Test developments

A logical basis on which to start is with military oil specifications. These incorporate the tests with which both the automotive and petroleum industries are best acquainted, since the procedures have been developed jointly by the two industries and the services. In addition, the Ordnance Group has indicated a strong desire to have the military specification conform to what is available on the civilian market, so as to simplify the supply problem in an emergency.

With this general approach in mind, the problem falls logically into three phases:

1. To determine what level of performance the current tests define in absolute terms.

2. To establish several ranges of test severity that would be suitable for all existing oils and also allow for improvement in lubricant quality.

3. To find out at what point in the range of test severity the oils meet current requirements.

Since the first abortive attempt to establish oils that could be used as standards of performance, several have been developed that appear to have the ability to react satisfactorily in more severe high-speed, shock-load tests, yet which are also satisfactory for lorries. T. P. Sands mixed one of these oils in various concentrations with base oil and set up a series of fifteen quality levels. Vehicle manufacturers were then invited to state their present and future requirements for high-speed, shock-load resistance.

Many tests are going on at present to determine this level. Originally, it was felt that it would be about 12, which is the level of oils with active sulphur additives. Since that time, however, several manufacturers have found that by phosphate-treating their gears, they can manage with levels of about 8, which appears to be that required by the services. Since the present level is about 5-6, it is obvious that future universal or multi-purpose gear oils will include much more high-speed activity additive.

Lubricant formulations

On the whole, the work of establishing laboratory tests to define more severe levels of operation has proceeded well. The field tests show that the requirements are at least beginning to be met but, owing to secrecy maintained because of competition between manufacturers, there is not much data available about the oils needed.

However, several oils have been reported that are apparently capable of passing the number 12 severity level, and which still pass the high torque test for wear. Recent work indicates that chemical corrosion may be of importance. Some improvement in the reduction of deposits has been obtained, but the trend towards increased axle temperatures means that this tendency will have to be watched closely. Other than this, the problems of introducing these oils apparently are economic rather than technical.

Combustion Chamber and Porting Development

A Survey of the Activities at the Laboratories of Weslake and Co. Ltd.

TWO aspects of the internal-combustion engine in which mathematics must give way to practical skill and experience are those of porting and combustion chamber design. It is axiomatic that the power output of an engine depends first on the efficiency with which the gases are introduced into and removed from the cylinders, and second on how well and at what rate it is burned. These two aspects are, of course, closely allied and while there are certain basic principles with which any competent designer is familiar, they represent a realm where the specialist comes into his own. In this country, among the leading firms who specialize in the phenomena of breathing and burning is Weslake & Co. Ltd., of Rye Harbour, Sussex. The company is led by Mr. Harry Weslake, A.M.I.Mech.E., M.S.A.E., whose experience of these subjects extends back well over 30 years.

As an indication of the sound basis of the firm's knowledge, it is apposite to mention some of the historical highlights which have brought Mr. Weslake to his present eminent position. He began, like many others, in the hard but effective school of motor-cycle racing and his first experiments on air flow were conducted in 1923 on Sunbeam motor-cycle engines. Flow testing revealed the reason for hitherto unexplained differences in the outputs of apparently identical units.

Much was learned during the 'twenties at Brooklands and elsewhere, and a major milestone was passed when Mr. Weslake was brought in as consultant to the original Bentley company, then in the heyday of its fame. He had a hand in the development of the six-cylinder 6½ litre power unit which, in the Speed Six model, won the Le Mans 24-hour races of 1929 and 1930. Following the Bentley successes, he was concerned with the Coventry Climax engine used in Triumph and Crossley cars, the overhead valve Jaguar and the 2 litre Armstrong Siddeley engines.

Post-war activities

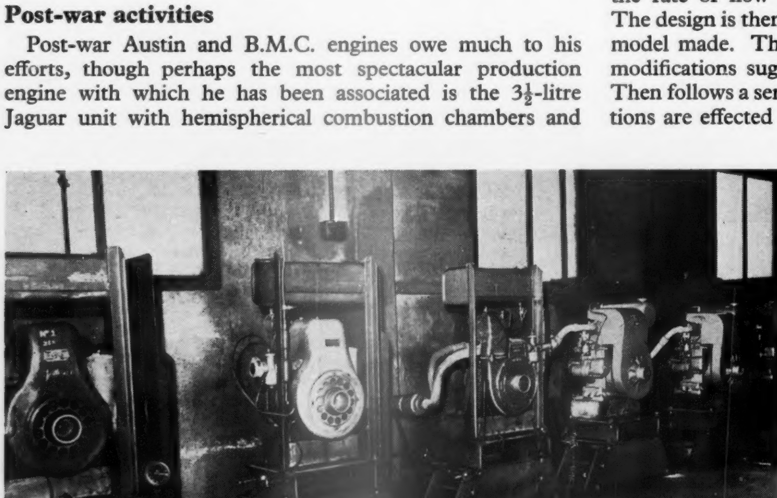
Post-war Austin and B.M.C. engines owe much to his efforts, though perhaps the most spectacular production engine with which he has been associated is the 3½-litre Jaguar unit with hemispherical combustion chambers and twin-overhead-camshaft actuation of the valves. Apart from its unusual combination of power and flexibility in standard form, this outstandingly successful engine has to its credit a vast number of competition victories, which include the Le Mans races of 1951, 1953, 1955 and 1956.

Other high-output engines which have received treatment are the racing Connaught, Vanwall and Coventry Climax units and the later twin overhead-camshaft Norton motor-cycle engines, on which Mr. Weslake collaborated with Joe Craig, then Nortons' development engineer. Daimler engines, particularly the 2½ litre six-cylinder unit, have undergone modification at the Weslake Laboratories and consultant work on various projects has been undertaken for the Ministry of Supply. The effects of highly leaded fuels on exhaust-valve life is a topical subject that has undergone considerable investigation.

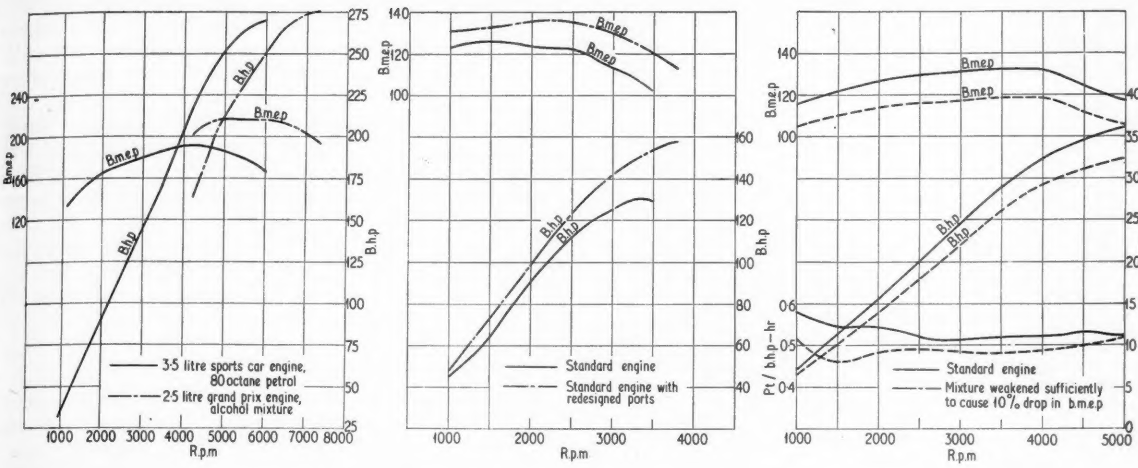
General principles

A guiding principle applied at the laboratories is that, so far as possible, any improvements made shall not be at the expense of disadvantages in other respects, whatever the duty of the engine under consideration. For example, an improvement in the breathing, which is not obtained by camshaft modifications or by increases in choke, port or valve sizes, is effective throughout the speed range. The improvement that can be obtained by attention to considerations relating to gas flow is shown clearly by one of the accompanying performance curves. Alteration to the form of the inlet port may increase turbulence to the benefit of low-speed torque, without loss at higher r.p.m.; and re-siting the sparking plug can improve starting or reduce a tendency to detonation, again without any detriment in other respects.

In the investigation of air-flow characteristics, a start is made with the original design of cylinder head, by ascertaining the rate of flow at various valve openings up to full lift. The design is then laid out in the drawing office and a wooden model made. This model may, even at this stage, embody modifications suggested by experience and the initial tests. Then follows a series of tests on the wooden model. Modifications are effected by removing material or building up with plasticine; they are continued until optimum results are obtained, after which the drawing is amended as necessary. A swirl meter is used to measure both the rate of flow and the degree of turbulence at various points in the combustion chamber.



The group of small industrial engines shown here is undergoing leaded fuel tests. Problems associated with the operation of this type of power unit with leaded fuels have received considerable attention in the Weslake Laboratories at Rye



Left: Comparative b.h.p. and b.m.e.p. curves of a $3\frac{1}{2}$ litre sports engine and a $2\frac{1}{2}$ litre racing engine. Centre: By porting modifications based on air-flow principles, performance can often be substantially improved throughout the speed range. Right: Weakening the mixture until the b.m.e.p. fell by 10 per cent effected a considerable economy in the fuel consumption of an engine with well designed combustion chambers

Normally, air-flow tests are carried out with the aid of a small gasometer, which in principle is identical with a conventional industrial type of gas holder. However, its mode of operation is different in that it draws air through the test piece, instead of blowing it through. This method of operation is obtained by causing the sliding portion of the gasometer to be raised by a weight attached to a cable passed over a system of pulleys; the flow is calculated from the time taken to pass a known volume. To eliminate human error, the stop watch is operated automatically by the sliding section.

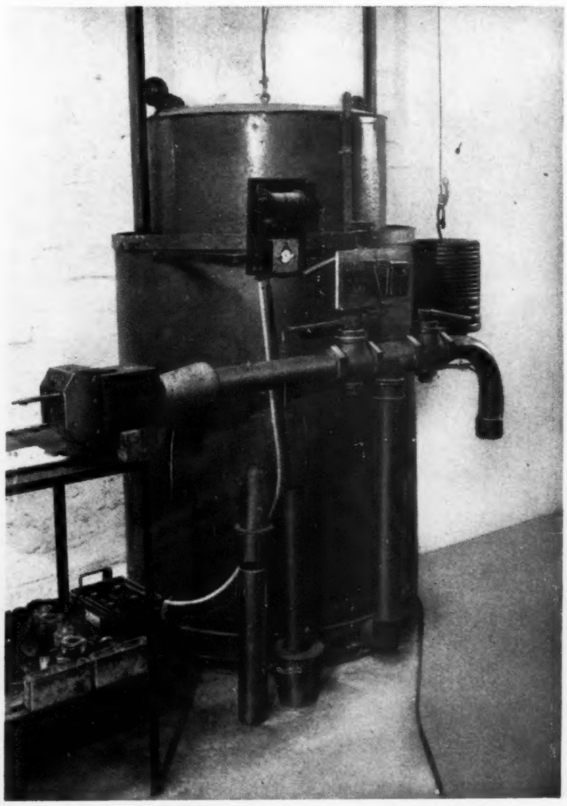
In experimental work for which the capacity of the gasometer is inadequate, a blower driven by an electric motor and discharging through an orifice is employed.

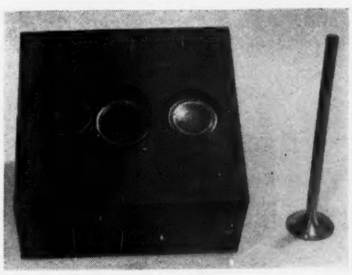
Engine testing

Apart from the air-flow testing, much running of complete engines is carried out on the seven Heenan and Froude water brakes in the laboratory. These are of a variety of sizes to cater for power outputs ranging from those of small car engines up to well over 500 b.h.p. The test-shop equipment includes items such as fuel-flow meters, indicators and a stroboscope for the accurate determination of ignition settings. Also, there is a Sunbury single-cylinder research engine, now modified in numerous respects to obtain an output higher than was originally intended.

In view of the potential advantages of the hemispherical or, more often, segmental-spherical combustion chamber for high performance units, much attention has been lavished on it. Except in motor-cycle form, it presents valve-actuation difficulties unless two camshafts are used, but it can enable excellent porting arrangements to be adopted, giving ample valve area and a relatively low octane requirement so that high compression ratios are possible. However, unless the inlet-port design is good, this layout suffers from lack of induction turbulence, with the result that the engine torque may be poor at lower speeds. In fact, on certain hemispherical-head racing engines with which the maximum advantage is taken of induction ramming, it has been proved that it is not resonance effects, but poor burning that has been responsible for the rapid falling-off in torque below certain engine speeds.

Although a degree of compression turbulence is obtainable by suitable design of the head and piston crown, it has the





Air-flow testing with the gasometer rig is shown on the right. The wooden model is of the Austin A35 head and is also shown to a larger scale

disadvantage, in comparison with induction turbulence, of being ineffective until the piston is nearing the top of its stroke. However, an adequate degree of turbulence can be imparted to the incoming mixture by attention to the inlet port. The incorporation of a port that is straight in plan view but inclined to the plane of the valve axes is one method of obtaining turbulence, but this is only practicable on single-cylinder units or those with one carburettor per cylinder, where the attachment flanges can be machined at the appropriate angle.

An alternative is for the tract to start parallel to, but offset from, the plane of the valve-axis, and then to curve round smoothly to the valve seat, so that rotational swirl is produced. With this arrangement, in which the valve stem is no longer in the middle of the port, the flow qualities are improved, and they may be further helped if the valve guide is assembled flush with the surface of the port. The offset and curved port can of course be applied to give directional swirl in heads other than hemispherical types.

Exhaust porting

On the exhaust side, because the direction of flow is the reverse of that through the inlet valve, different conditions obtain. A long, divergent port gives the best results and the projection of the valve guide and its boss are not considered to be critical from the point of view of air flow, though streamlining the boss is beneficial. Maximum cooling of the exhaust guide is obviously desirable, to keep down the temperature of the valve, and in this respect a wet guide—with ample coolant passages round it—has given favourable results. The Aston Martin engine is an example of the wet guide type of layout.

Considerable improvement has been effected on a variety of engines by the change from siamesed to separate inlet ports. With siamesed ports, not only does one cylinder tend to rob its fellow of part of its charge, but the flow rate falls off more rapidly with increasing speed than with single ports. In the case of exhaust porting, siamesing has serious disadvantages: valve temperatures are higher and there is a reduction in the kinetic effect, with adverse effects on the scavenging of the engine cylinders.

Valves and seats

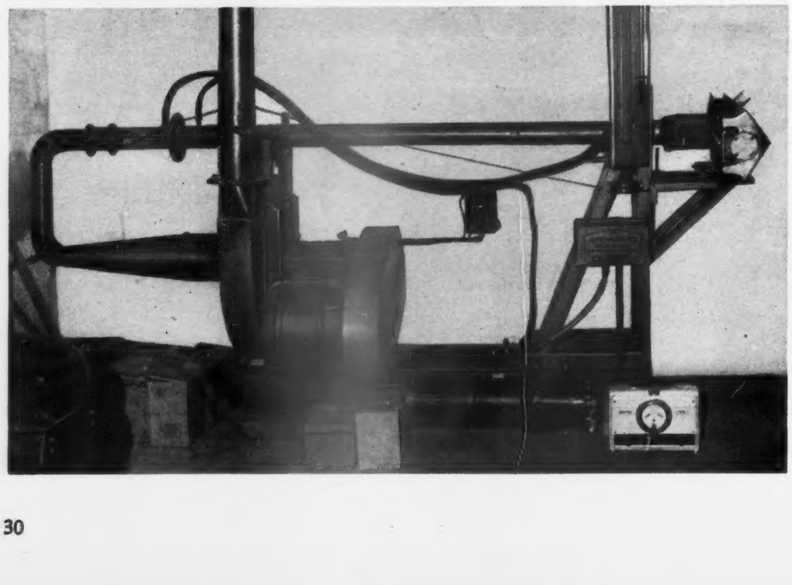
Flow through the inlet-valve opening at smaller lifts is appreciably aided by radiusing the bore of the valve seat ring to blend into the streamline, or venturi, shape of the port. A 30 deg valve seat shows to advantage up to about one-eighth of the total lift, in comparison with the more common 45 deg seat. However, the 30 deg angle is likely to be less effective if space round the valve is restricted, and it is not practicable with the spherical-back valve, which has proved its worth in the search for maximum output from hemispherical-head engines. With this type of valve the curvature of the fillet between the stem and head is convex, instead of concave, adjacent to the seating face, the radius of curvature of this spherical back being the same as that of the combustion chamber; also, the concave transition radius between it and the stem is kept to the minimum consistent with mechanical strength. For exhaust valves the 45 deg seat angle is the most suitable. It provides a more robust valve head and, because of its greater wedging effect, better self-cleaning properties than the 30 deg seat.

The manifolds

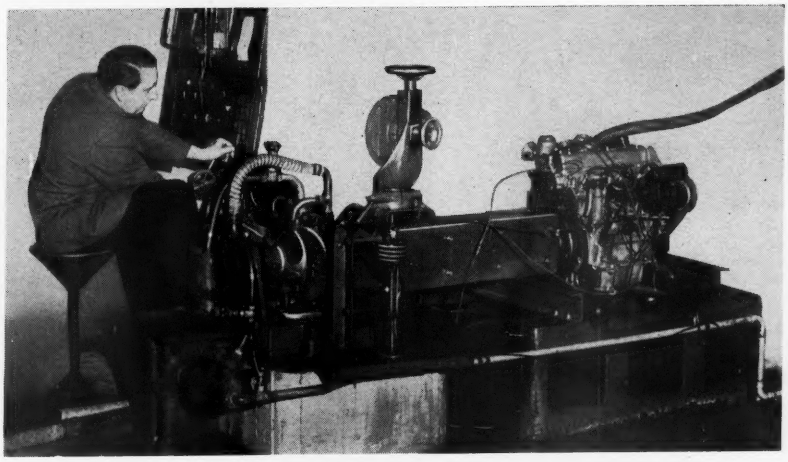
Manifold design has, of course, a considerable influence on the distribution of mixture to the various cylinders. If one cylinder gets a weaker charge than its fellows, the mixture setting has to be richer than if distribution were uniform, in order to avoid overheating or detonation in the cylinder containing the weaker mixture. It is Mr. Weslake's contention, though, that incorrect burning can greatly magnify the effects of unevenness in the distribution. If the combustion characteristics of an engine are satisfactory, the unit is much less sensitive to variations in mixture strength than one which is, say, deficient in turbulence.

One of the accompanying sets of curves illustrates the beneficial effect of good burning on fuel consumption. With

The Jaguar cylinder head mounted for test on the blower rig



Air-flow investigations are carried out with a blower rig as well as with the gasometer. On this rig is seen the cylinder head from a 3½ litre Jaguar sports-racing engine



Heenan and Froude water brakes are employed for tests of complete engines. Here a power run is being carried out on a Coventry Climax 1,100 cm³ power unit

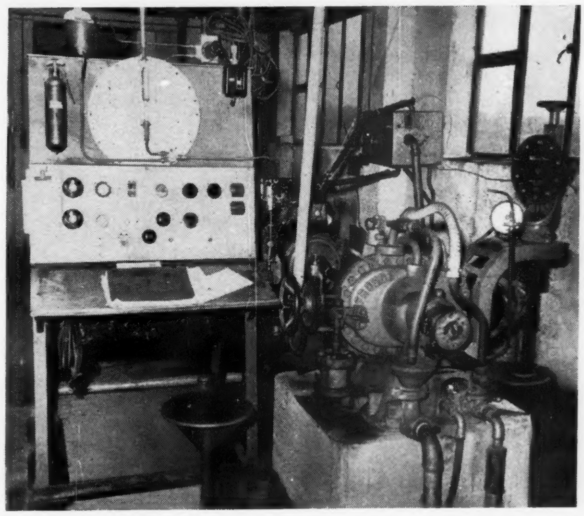
the engine concerned, which employs controlled swirl turbulence and a small amount of squish, a considerable fuel saving could be effected by weakening the mixture until a 10 per cent drop in b.m.e.p. was observed. It can be seen that between 1,100 and 4,500 r.p.m., on the weaker setting, the specific consumption remained below 0·5 pints/b.h.p-hr, with an optimum value of 0·46 pints.

Requirements for combustion

For efficient burning, the fundamental requirements are considered to be an open combustion-chamber layout with adequate swirl and with the squish arranged to project the charge at the sparking plug. The exhaust valve should be free and not too close to the inlet valve; if the design is such that the valves are unavoidably close together, pre-heating of the charge by the valve can be avoided by suitable profiling of the combustion chamber or by bias of the inlet port. It may be necessary to compromise a little on valve masking by the chamber wall in order to get the desired swirl away from the exhaust valve. Possibly, development may be with advantage directed towards the greater use of the piston crown to give the requisite combustion-chamber form.

It must not be thought from the foregoing that, because of the advantages of the hemispherical layout, other combustion-chamber forms have been neglected. Parallel disposition of the overhead valves is still by far the most common, and numerous Weslake patents have been taken out covering this layout. One of the most successful designs has been developed from the familiar bath tub chamber, to embody the features just enumerated. In plan view the chamber is of heart formation with a pronounced peak in the wall, extending into the space between the valves. This peak provides directional control of the incoming mixture stream, keeping it away from the hot exhaust valve and promoting turbulence. In addition, the portion of the wall round the inlet valve remote from the cylinder axis is undercut round an arc of about 180 deg, so that the charge is directed across the chamber towards the sparking plug. The plug is sited remote from the valves but its exact position is not defined in the patent. It is chosen by trial and error to give the best combustion under lean-mixture, part-load conditions.

From the search for improved cold starting and behaviour during warming up, it has been concluded that the water-jacketed inlet manifold is a better compromise than the hot spot. This is because the hot spot has the disadvantages that its effect is influenced unduly by the ambient temperature and by the driving methods employed; in conditions of hot weather or hard driving, the charge density may well be excessively reduced, with adverse effects on performance.



Instruments and brake details for the testing of an Austin A90 engine

An inevitable difficulty in connection with the high compression ratios possible with 100 octane fuels is that of obtaining a satisfactory combustion chamber shape. This problem is accentuated by the continuing tendency towards larger bore-stroke ratios. When such ratios much exceed unity and the large valves thus permitted are employed, the higher the compression ratio the more difficult it becomes to avoid an elongated, shallow chamber with restricted valve lift.

It is in this respect, rather than the elimination of valve springs, that desmodromic valve operation can be of such benefit on racing and sports-racing engines. With this type of actuation, valve acceleration and deceleration rates can be increased so much that overlap can be appreciably reduced and an adequate lift provided without risk of contact between valve and piston.

Although Mr. Weslake is reasonably satisfied with the progress made on filling and burning techniques, he is fully aware that much remains to be learned. At times, there is justification for criticism of engine designs in which efficiency in these matters is sacrificed to production expediency. Close liaison between designer and production engineer, to minimize the adverse effects of any compromises while keeping costs as low as possible, is an extremely important requirement that, in some instances, is not fully appreciated.

Renold Auto-Adjuster

The Type S.C.D. Chain Tensioner, Which is Adjusted Automatically as Wear Occurs

THE desirability of providing for automatic adjustment of chain-type camshaft drives is, of course, well known to engine designers. Automatic adjustment, if correctly applied, contributes materially towards mechanical quietness, reduces wear and ensures precision of timing. An essential feature of correct adjuster design is a non-return mechanism, that is, the adjuster must be capable of automatically taking up new positions as the chain wears. Also, it must permit a limited reverse movement, sufficient only to prevent over-tensioning and to allow for damping so that the greatest practicable degree of smoothness of operation is obtained. The solution of this problem is difficult because of space restrictions and the need to keep prime costs as low as possible. These problems have been solved in an ingenious manner in the Renold type S.C.D. Auto-Adjuster, which has an internal restraint mechanism that is noteworthy for both its compactness and simplicity.

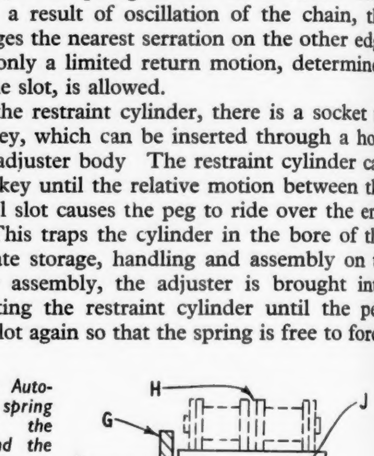
As can be seen from the accompanying illustrations, the main component of the assembly is a hollow plunger, on one end of which is carried a rubber head, which bears against the chain. The rubber composition and shape of the head that give optimum performance have been determined as a result of extensive research. The shank of this plunger is carried in a cylindrical bore in the body of the unit, which is bolted to the front wall of the engine crankcase. A restraint cylinder is housed in the hollow plunger, and a compression spring is interposed between this restraint cylinder and the end of the plunger on which the head is carried. The function of this spring is to force the two components apart, the restraint cylinder bearing against the closed end of the bore in the housing, while the rubber-faced head bears against the chain.

A helical slot is machined in the periphery of the restraint cylinder. The edge of this slot that is nearest to the head is smooth, while the other edge is serrated. A peg, carried in the plunger, projects radially inwards into the bore and registers in this slot. When the plunger moves outwards relative to the body, this peg rides over the smooth edge of the slot. However, if the plunger tends to be forced back into the body, as a result of oscillation of the chain, the restraint peg engages the nearest serration on the other edge of the slot; thus, only a limited return motion, determined by the width of the slot, is allowed.

In the base of the restraint cylinder, there is a socket to receive an Allen key, which can be inserted through a hole in the end of the adjuster body The restraint cylinder can be rotated by the key until the relative motion between the peg and the helical slot causes the peg to ride over the end of the cylinder. This traps the cylinder in the bore of the plunger, to facilitate storage, handling and assembly on to the engine. After assembly, the adjuster is brought into operation by rotating the restraint cylinder until the peg enters the helical slot again so that the spring is free to force the headed plunger outwards against the chain in the manner shown in the accompanying illustrations.

Cross section of the Auto-Adjuster, showing the spring interposed between the restraint cylinder and the head of the plunger

A Oil inlet hole in spigot
B 0·250 in diameter bolt
C Spigot
D 0·250 in diameter bolt
E Body
F Rubber head
G Back plate
H Duplex roller chain
J Oil outlet hole
K Spring
L Plunger with end piece
M Cylinder
N Limit peg
P Socket for Allen key
Q Plug



The unit is fed with oil, under pressure, for the lubrication of both the adjuster and the chain and also to damp the action of the adjuster. Oil enters the unit through a hole in a spigot on the adjuster body and leaves through a metering orifice in a plate in the headed end of the plunger. From this orifice, the lubricant passes through a hole in the rubber head and thence is directed on to the chain. The spigot, which is on the face of the body that seats on the engine crankcase, registers in a hole drilled into the front wall of the crankcase, to communicate with one of the ducts of the engine lubrication system. Different versions of the adjuster are supplied, according to whether continuous or intermittent flow lubrication is required.

Since the adjuster incorporates a positive return stop, in the form of the serrated edge of the slot in the restraint cylinder, only a relatively light spring pressure is necessary. This, together with the fact that the unit is well lubricated and damped, reduces wear to a minimum. The design is compact and economical and is suitable for most automobile engines. Setting and operation are automatic, so assembly is simple and attention is not needed during service. Noise is effectively reduced throughout the whole life of the chain. Tolerances, with regard to centre-to-centre spacing of the wheels, are automatically accommodated, and it is not necessary to incorporate any additional device for the lubrication of the timing drive when the engine is running.

Timing drive assembly incorporating the Renold Type S.C.D. chain tensioner. A noteworthy feature is its compactness

THE TAREX AUTOMATIC TURRET LATHE

A Swiss Single-spindle Machine for either Continuous or Batch Production of Bar or Chuck Work

A CHARACTERISTIC of production in the motor industry is its adaptability. This quality was amply demonstrated during the war period when, often on extremely short notice, it switched its productive capacity to new or unfamiliar work to meet current and changing requirements. Less drastic changeovers are called for under what may relatively be termed "normal" conditions, in order to keep abreast with the advancements in design and the fashion trends which are features of such a highly competitive industry. Quantitatively, the most carefully arranged production plans are likely to be bedevilled by the summary imposition, either at home or abroad or both, of prohibitions, restrictions, quotas or taxes on motor vehicles in furtherance of national political, economic or fiscal policies. It is desirable, therefore, that production equipment should be capable of operating with efficiency at different rates of production and of handling different types of work so that heavy investments of capital may be avoided in the event of a change in production.

The single-spindle automatic lathe, Fig. 1, produced by Tarex S.A., Geneva, Switzerland, was designed to meet such conditions. It is a high precision tool which may be classed as a logical development of the bar repetition or the chucking automatic lathe. Embodying the technique of the screw auto, it has a universal spindle for either bar or chucking work, faster and more comprehensive chucking arrangements, and a six- or eight-position turret. Both the front and the rear slides are compounded. On the front slide transverse and longitudinal movements are each cam-controlled but on the back slide only the transverse movement is cam-operated and longitudinal movement is controlled either by turret advance or by a template. The vertical slide is adjustable longitudinally.

Movements of the turret and the slides—except in the possible instance of the back slide, previously noted—are completely independent. Slides are operated by standard flat cams and adjustable-ratio levers. An auxiliary shaft and accelerator gear is provided to minimize idle time. Additionally, a fourth slide or a thread-chasing attachment can be fitted and independently operated. Accessory equipment for such operations as high-speed drilling and multiple drilling and tapping, can be driven by an independent motor and bevel gearing through the hollow turret shank, or side drilling and tapping by a motor on the front or back slide.

The aim has been to produce what may be classed as a universal machine. Equipped with automatic loading and unloading equipment, it can be fitted into a transfer line, while for batch production it can be used as a special-purpose machine and requires only a reasonable investment for new tooling at a changeover in production. Re-setting can be effected in the minimum of time and all slide movements are obtained with the standard cams. A special cam is required only for the turret slide, and this can be utilized for a variety of components having similar operational sequences. That its use enables production to be speeded-up with reduced manpower, and more complicated work to be produced by unskilled labour, has been well appreciated by the European motor industry which, to date, has absorbed approximately 80 per cent of the machines produced.

In this field installations have tended to fall into two main application groups. The first group is for precision finishing operations on components that have previously been rough-machined or part-machined. The second is for roughing and finishing operations to machine a component as completely as possible in one cycle and with one setting.

With six or eight turret positions, up to four slides (two of which have compound movements), and the available wide range of independent attachments such as hydraulic

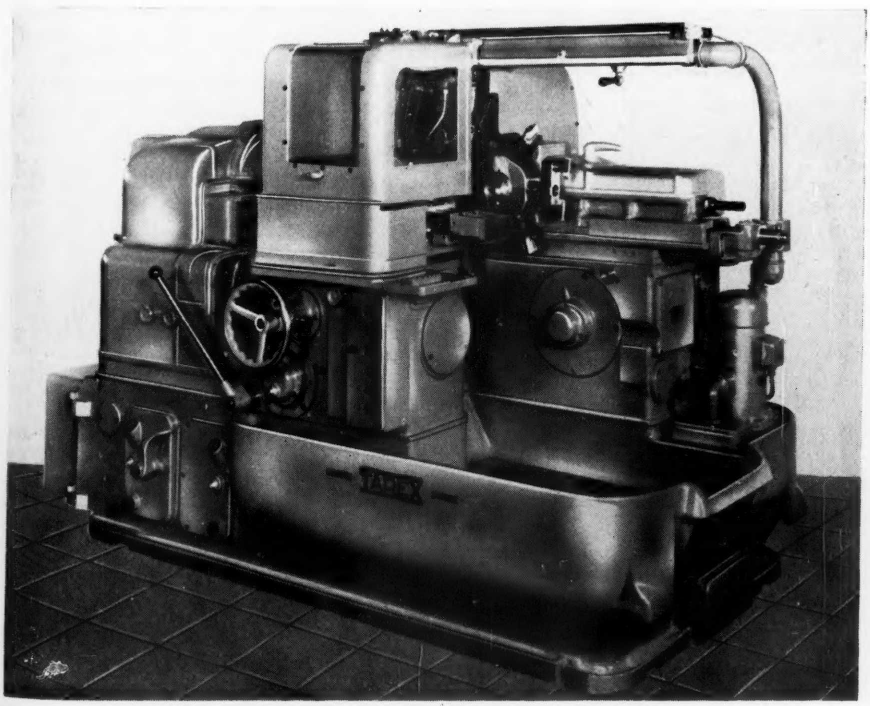


Fig. 1. The Tarex single-spindle automatic turret lathe, model TAR-H

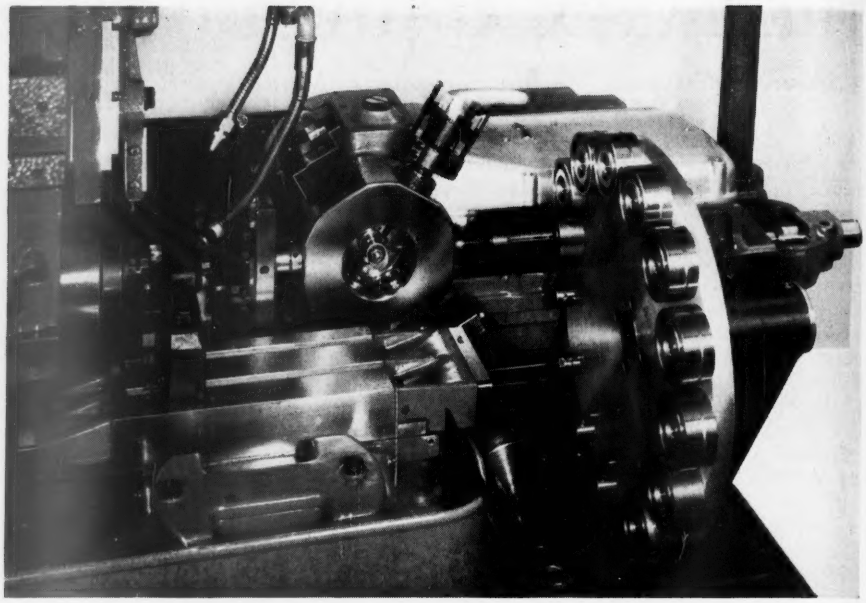
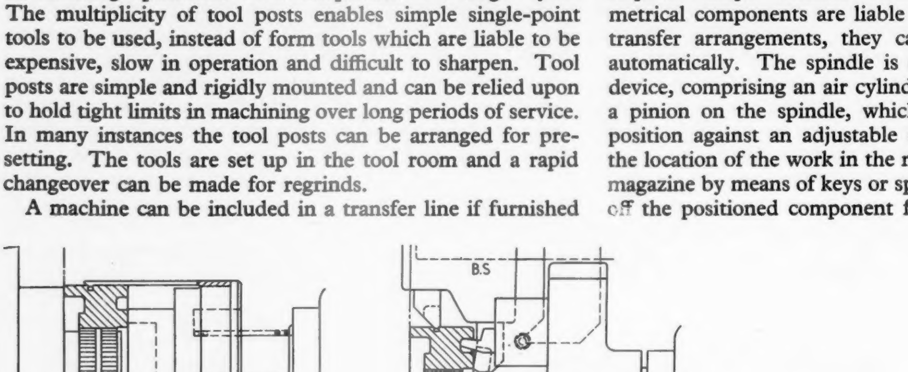


Fig. 2. Finishing synchronizer bodies on the Tarex machine. Located parts are automatically loaded from a drum-type magazine and automatically unloaded

copying, thread chasing, multi-drilling, multi-tapping, and side drilling, the machine is capable of combining most or all finishing operations on a component in a single cycle. The multiplicity of tool posts enables simple single-point tools to be used, instead of form tools which are liable to be expensive, slow in operation and difficult to sharpen. Tool posts are simple and rigidly mounted and can be relied upon to hold tight limits in machining over long periods of service. In many instances the tool posts can be arranged for presetting. The tools are set up in the tool room and a rapid changeover can be made for regrinds.

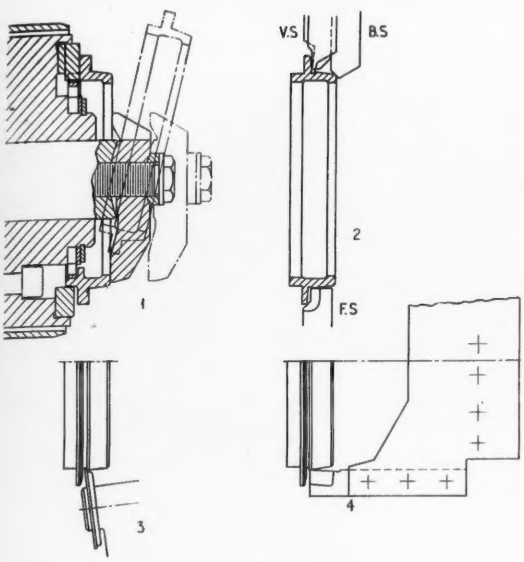
A machine can be included in a transfer line if furnished with a magazine feed and automatic loading and unloading devices. In the case of symmetrical components, it then only requires simple transfer mechanisms. Although unsymmetrical components are liable to complicate somewhat the transfer arrangements, they can be satisfactorily handled automatically. The spindle is equipped with a positioning device, comprising an air cylinder operating a rack engaging a pinion on the spindle, which is brought to the desired position against an adjustable dead stop. Also involved is the location of the work in the receivers of, say, a drum-type magazine by means of keys or splines. The loader then picks off the positioned component from the magazine, transfers



Op.	Cut m/min	Speed r.p.m.	Stroke mm	Feed mm/rev
1	Auto-	load		
2	104	671 R	2	0·1
3	160	671 R	(2)	(0·1)
4	135	671 R	(6)	(0·15)
			(0·5)	(0·03)
4 F.S	160	671 R	35	0·2
4 B.S	160	671 R	(6)	(0·2)
			(6)	(0·5)
			(10)	(0·2)
	Ret.		(10)	(0·2)
			(6)	(0·5)
			(6)	(0·2)
5	90	450 L	1	0·035
5 V.S	107	450 L	(2)	(0·15)
			(0·6)	(0·03)
6	Auto-	unload		

B.S—back slide, F.S.—front slide, V.S—vertical slide

Fig. 3. Tooling layout for steel synchronizer body. Outside diameter 75·2 mm



Op.	Cut m/min	Speed r.p.m.	Stroke mm	Feed mm/rev
1	Hand-	load		
2 B.S	215	530 R	5	0·08
2 V.S	200	530 R	5	0·08
			1·5	0·05
2 F.S	180	530 R	13	0·10
3	132	380 L	12	
3 V.S	200	530 R	1	0·07
	Slow-	ret.	5	0·20
4	180	530 R	2	0·05

B.S—back slide, F.S—front slide, V.S—vertical slide

Fig. 4. Tooling for finishing synchromesh cone. Brass stamping, outside diameter 119·45 mm

it to the machine, and the positioned spindle takes it from the loader. The machine can thus match current trends in production and enables the advantages of transfer and automaticity to be realized on the smaller turned components. Applied in this manner for a cycle of finishing operations it is normally used for the production of from 400 to 1,200 components per two-shift day.

When the machine is used for completely machining components, the number of tool posts and standard attachments are again of advantage in enabling a large number of operations to be arranged in one cycle. Its rigid construction makes it suitable for roughing as well as for finishing operations and ensures reliability and the consistent maintenance of accuracy. The short changeover and setting times make it suitable for the batching of components in relatively short runs at more modest rates of production, say from 100 to 400 per two-shift day.

One European manufacturer finds it advantageous to turn one end of a complete batch of components and then to change the tooling and reset to turn the other side of the work. Usually several sets of tools for producing different components are held available for current use. It is the common practice of the firm to supply with a machine several sets of tooling and equipment which are manufactured at the Geneva works. The tools are thoroughly tested on the machine and production rates are proved before the machine is dispatched. A service engineer is freely available to demonstrate the changeover and to instruct the machine setter.

Design features

On the basic foundation of a rigidly constructed bed and base, efficient operation of automatic movements depends on careful fitting and mounting of the various mechanisms. On the Tarex machine these are grouped in sub-assemblies which are fitted on the bench and subsequently dowelled and bolted in position. They can readily be removed as units for adjustment or maintenance.

Headstock

All headstock gears are hardened and ground and are mounted on short, stiff shafts running in roller and ball bearings. Cone clutches are used, as they have been found more positive in action than units of other design. All four clutches can be adjusted for wear by means of a single wrench. Automatic reversing is positive and accurate in operation and plain taps can be used for blind holes, and button dies can be run up to shoulders.

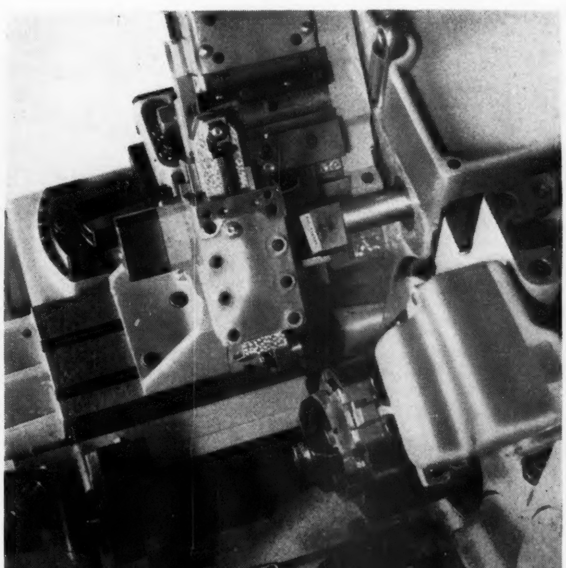
Each pair of change gears provides two forward speeds and two reverse speeds. These range from the extremes of 2,551 r.p.m. in reverse to 60 r.p.m. in the normal forward direction. With the standard change gears 67 machine cycles are available, the fastest being 28 seconds and the slowest 800 seconds. To reduce unproductive time, the camshaft speed can be increased during any cycle by an automatic accelerator gear. The accelerated speed is 40 seconds for one complete camshaft revolution.

Spindle

Although the accuracy of components produced on the Tarex machine is guaranteed to 0·0005 in, the spindle is not individually fitted, but is precisely made and is interchangeable. It is designed as a universal spindle capable of accommodating, without modification, an extremely wide range of work-holding equipment. It can be fitted with collets; two- or three-jaw chucks, either mechanically, pneumatically or hydraulically operated; expanding mandrels; or hollow chucks, to handle black bar or to accommodate a tool through the spindle in order to save a follow-on operation.

Special duplex chucks with two sets of jaws each operated by an independent air cylinder can also be fitted. These provide high-pressure holding for rough machining and a lower-pressure holding for finishing cuts. By this means it is possible to avoid the distortion of relatively thin-section components. Collets are seated against the ground spindle

Fig. 5. Thread chasing attachment with copying slide set up for cutting thread on synchromesh cone



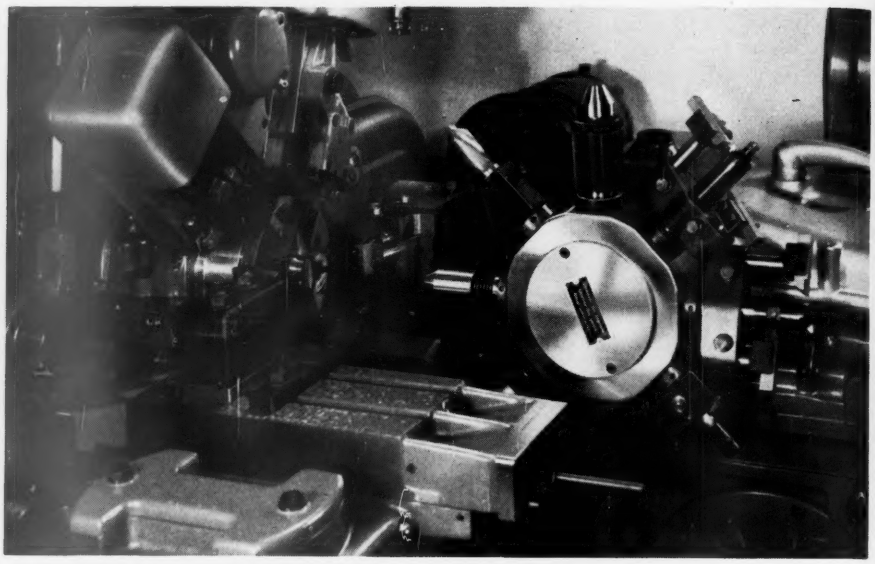


Fig. 6. In this set-up a water pump body is multi-drilled, 3 holes, from the turret and side-drilled, 2 holes, from the back slide

cap and are opened and closed without longitudinal movement. Longitudinal limits can thus be held on the work irrespective of the commercial tolerances of the bar stock. The thrust developed by the chucking levers can be reversed and applied to plain or hollow toggle chucks.

The spindle is fitted with a brake to reduce unproductive time. When the machine is operated as a chucker, the feeding mechanism can be used for the positive ejection of finished work. Pneumatic chucks are operated by an air cylinder mounted at the rear of the spindle and, as a safety measure, should the air pressure fall below 60 lb/in^2 the lathe is automatically stopped. Synchronization of the feeding and chucking operations with the other automatic movements of the cycle is accomplished by trip dogs located on the timing drum.

Turret

The turret slide is actuated by two separate and independent movements, advancing to and withdrawing from the work, which are cam operated. Turret indexing is under the combined action of a crank and a Geneva-cross mechanism, and is synchronized with the lobes of the cam by trip dogs on the timing drum. The flat cams are cut from steel plates and hardened. Tool holders have $1\frac{1}{4}$ in diameter shanks and are locked in position in the turret holes by means of patented concentric locking bushings. These bushings ensure accurate centring with no cross strain and cannot mar the shank and impair the fit, as is possible with conventional cross-locking devices. One turret position is usually allocated for the stock stop. This position can be released for tooling if desired by the addition of a swing stop, which can be supplied and mounted as a standard accessory without the need for special fitting. The swing-stop attachment synchronizes automatically with the feeding mechanism.

Front slide

The compound front slide is arranged with the transverse motion below the longitudinal motion. The cross motion is

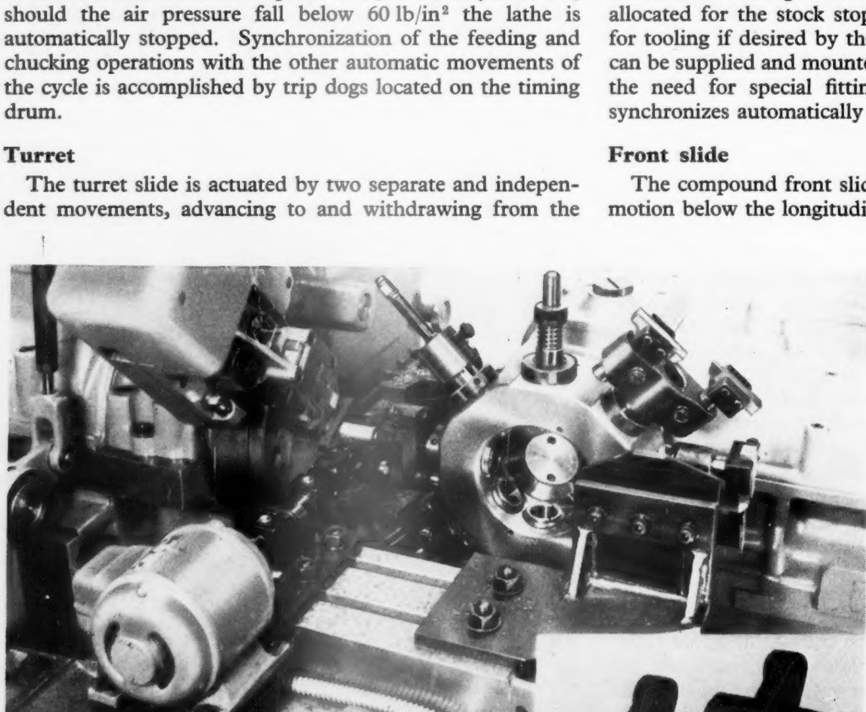


Fig. 7. A chain tensioner housing is machined from both ends in one cycle. Reversing of the component is by a motor-driven attachment on the front slide

transmitted from the cam to the tool through a solid lever with a toothed segment engaging the rack on the slide, while the longitudinal movement is provided by another cam through a lever with arm ratios that are adjustable between 1 : 1 and 1 : 2·5.

Back slide

The back slide is also compounded and is similar in design to the front slide. In this case also, the cross movement is operated by cam and a lever with arm ratios adjustable between 1 : 1 and 1 : 2·5. Longitudinal motion, however, is actuated from the turret or from a template. As in the case of the front slide, accurate tool setting is accomplished with a micrometer setting screw.

Vertical slide

This slide is inclined to the rear at 20 deg from the vertical, in order to allow more space for the standard attachments that can be fitted between the front and the vertical slides. Cross feed is from a cam with an adjustable lever having ratios between 1 : 1 and 1 : 2. By the use of appropriate tool holders, the slide can be operated equally well at either direction of spindle rotation. The whole slide can be displaced parallel to the spindle axis to position the tool accurately in relation to the spindle nose. Spacers to a maximum thickness of 2 in can be inserted between the tool holder and the slide to advance the tool still further for chamfering or similar light operations.

Fourth slide

This is supplied as an attachment which can be mounted without fitting on a prepared faced on the headstock. Located between the front and the vertical slides, it is used for chamfering, grooving and similar operations.

Cams

All the four slides are controlled by flat cams, independent of each other. The cams are not keyed to their shafts but engage with the driving pin of a 100-tooth clutch. Their location on the shaft can thus be adjusted to ensure synchronization of the slides by increments of 0·01 of the cycle. Since the position of the cam is adjustable in this manner, and as the stroke of the tool can be varied by means of the adjustable ratio lever, it becomes unnecessary to cut a slide cam for each workpiece. The desired movements of the tools can be achieved with the standard cams supplied with the machine.

The transverse movement of the front slide will seldom need adjustment of stroke or feed. It is, however, necessary to control the dwell period when the tool is in its forward position. This is readily set by means of a pair of fan cams interlocking by means of face serrations. The concentric portion of the cam can be increased or reduced by increments of 0·01 of the cycle.

Tooling Layouts

Four representative tooling layouts, all actual set-ups

Op.	Cut m/min	Speed r.p.m.	Stroke mm	Feed mm/rev
0	Hand-	load		
1	18	330 L	7	0·25
2 B.S	86	330 L	29	0·15
			3	0·06
2	15/25	330 L	64	0·25
			2	0·06
3	58	1159 L	65	0·25
4	17	330 L	65	1·00
5A	Rever	se		
5B	Auto-	load		
6	95	1159 L	13	0·15
			2	0·05
7 4.S	26	330 L	(2)	(0·05)
7	25	330 L	3	0·06
8	10	140 R		

B.S—back slide, 4.S—4th slide

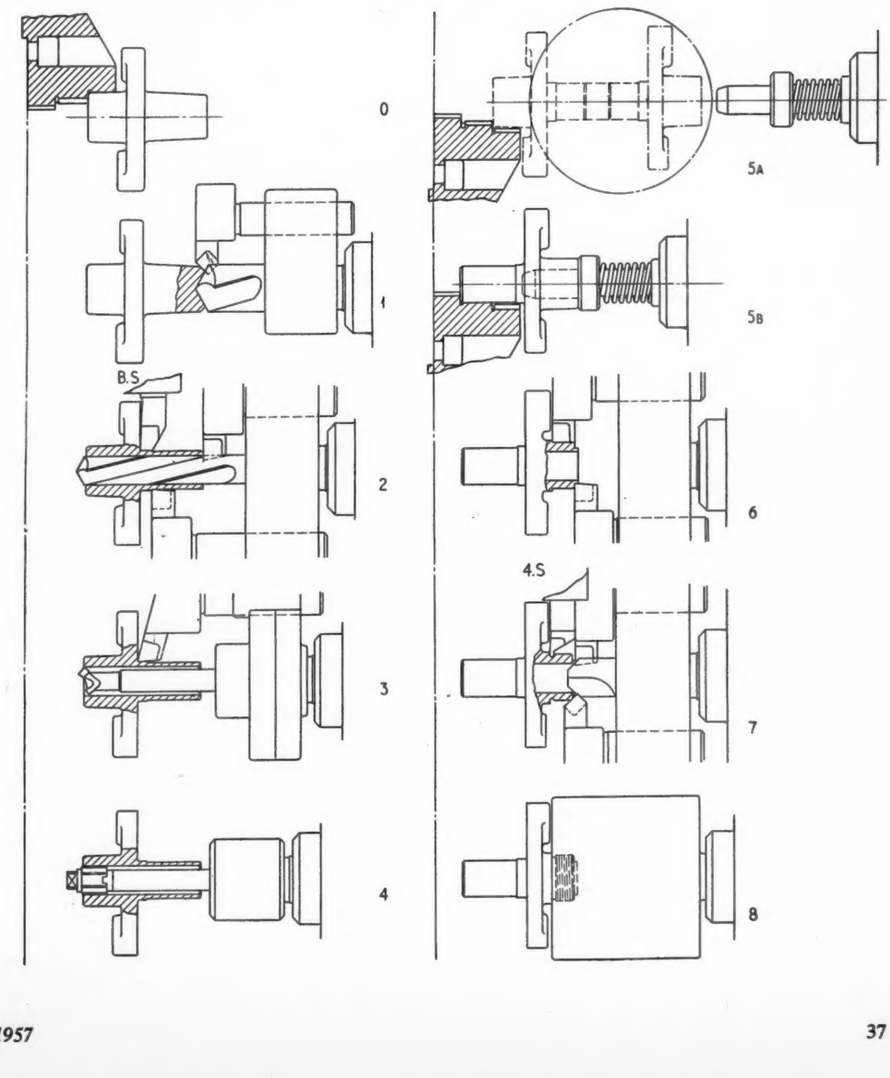
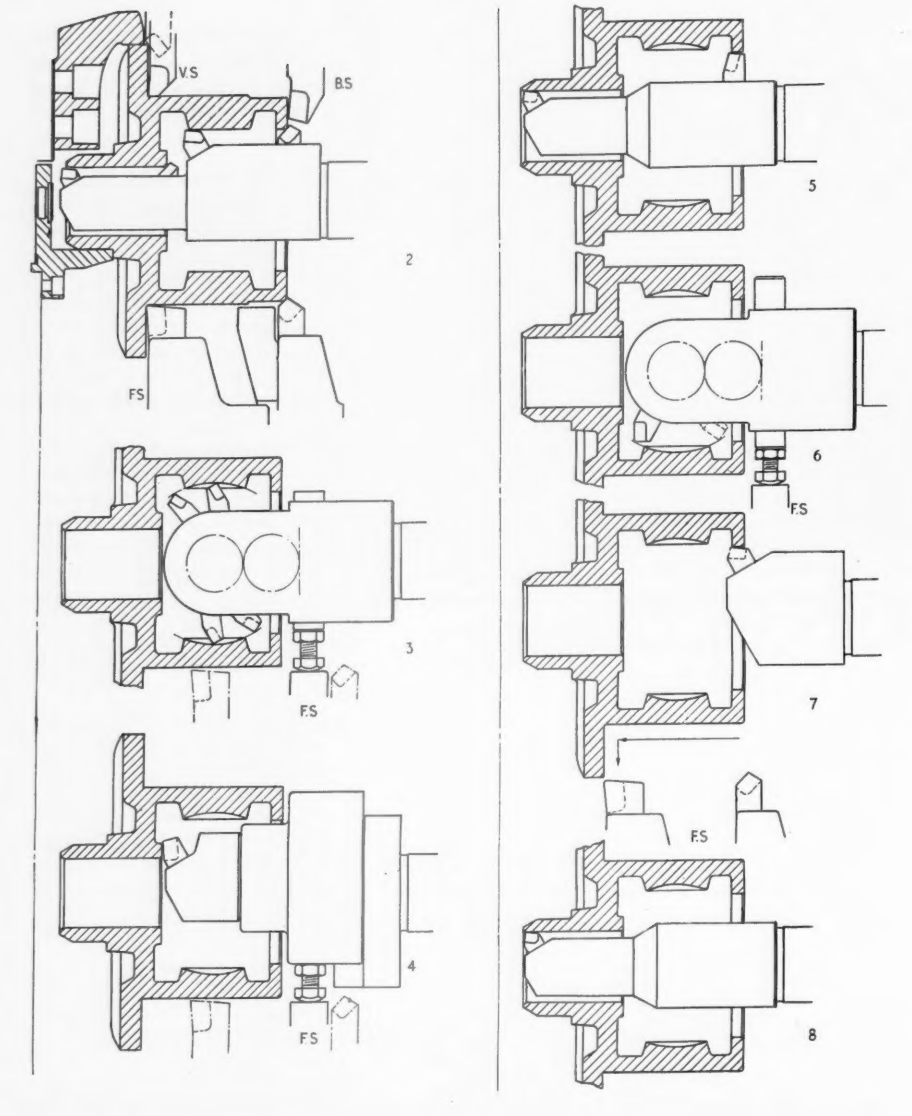


Fig. 8. Operational sequence for machining both ends of cast iron chain tensioner housing, with automatic reverse and re-load



Op.	Cut m/ min	Speed r.p.m.	Stroke mm	Feed mm/ rev
1	Hand-	load		
2	54	227 L	54	0·3
			2	0·1
2 V.S	120	227 L	28	0·15
2 B.S	80	227 L	18	0·25
				0·15
2 F.S	80	227 L	55	0·3
3	62	227 L	18	0·15
4	57	339 R	10	0·15
5	40/80	339 R	54	0·3
6	90	339 R	36	0·12
7	80	339 R	8	0·12
7 F.S.	80	227 L	76	0·15
	120	227 L	28	0·15
8	93	796 L	54	0·12

B.S—back slide, F.S—front slide, V.S—vertical slide

Fig. 9. Layout for malleable iron differential gear housing. Diameter over flange 165 mm

used by European motor manufacturers, serve to indicate, but not to exhaust, the versatility of the Tarex lathe. Fig. 3 shows the application for finishing the body of a synchromesh mechanism. In order to achieve the necessary concentricity and squareness with the previously splined bore, the component is chucked internally on the splines.

The operational sequence is as follows:

1. Automatic loading
2. Chamfer on two diameters
3. Chamfer on two diameters
4. Turn recess on two diameters
4. (front slide) Turn outside diameter
4. (back slide) Face; chamfer O.D. groove
5. Form groove in recess
5. (vertical slide) Finish O.D. groove
6. Automatic unloading

Movements shown bracketed on stroke and feed in the tabulated data alongside the drawing are overlap movements that do not add to the cycle time. Since the work is chucked on the splined bore, it necessitates the automatic positioning of both the spindle and the component to permit automatic loading. The set-up of the machine for this component, complete with drum-type loading magazine, is shown in Fig. 2. Production time is 33 seconds.

Another finishing application is shown in the tooling layout, Fig. 4, for a brass synchromesh cone. To avoid the possibility of distortion, special arrangements are made to clamp the component to the spindle and hand loading is employed. The part is radially and axially located in the holding device and is secured by a three-armed clamp.

Operations are sequenced as follows:

1. Load by hand
2. (back slide) Copy turn 4 deg flange taper; chamber edge of cone
2. (vertical slide) Face flange; form groove
2. (front slide) Copy turn 7 deg cone; chamfer edge of flange
3. (thread chaser) Cut thread on cone
3. (vertical slide) Finish groove
4. Finish turn cone max. diameter

Taper turning is possible as both front and back slides have compound movements. The thread chaser is an independent attachment with its own copying slide and is shown in the set-up, Fig. 5. Production time is 56 seconds.

The layout, Fig. 8, for a cast iron chain-tensioner housing is of interest as the component is machined on both sides in a single cycle and is automatically reversed in the chuck after one side has been completed. This operation is effected by a small motor mounted on the front slide, as will be seen in the

illustration of the complete set-up, Fig. 7. Machined from the rough casting, the production time is 161+6 seconds; the sequence being as follows:

0. Hand loaded, at zero position
1. Centre drill; chamfer end
2. (back slide) Face flange
2. Drill; turn hub; face end
3. Bore; turn hub root
4. Ream bore
5A. Automatically reverse work
5B. Automatically reload work
6. Turn shank on two diameters
7. (4th slide) Form groove
7. Chamfer end, countersink bore, face end
8. Thread shank with turret die-head

An example of the larger and heavier class of component that can be handled on the Tarex machine, for roughing and finishing in one cycle, is shown in the layout, Fig. 9. This is for a differential gear housing in malleable iron and measuring 165 mm diameter over the flange. Spherical turning tools, operated through a rack and pinion by the front slide, are required for machining the two internal part-spherical seatings. It will be noted that the roughing tools are duplicated; the individual tools each traverse only half of the seatings. The tolerance on the finish spherical diameter is 0·035 mm.

The sequential operations are:

1. Hand loading
2. Rough bore hub and body; countersink
2. (vertical slide) Face flange; chamfer
2. (back slide) Face end of body
2. (front slide) Rough turn body O.D.; chamfer end
3. Rough bore two part-spherical seatings
4. Face end of hub
5. Semi-finish bore hub and body
6. Finish bore part-spherical seatings
7. Finish bore body
7. (front slide) Finish turn body O.D.; chamfer end of body; finish face flange
8. Finish bore hub

Total production time is 580 seconds+10 seconds for loading and unloading.

Another interesting set-up is shown in Fig. 6. With this a cast iron water pump body is fully machined from the casting, the main bore being machined in three stages and reamed to a tolerance of 0·025 mm. When facing the three-armed flange from the vertical slide, the component is supported by a turret-mounted rotating steady centre. In the final operation three holes are drilled in the flange by a multi-drill unit on the turret and two holes are side-drilled at an angle in the body by a motor-driven multi-drill unit on the back slide.

All the illustrations and tooling layouts relate to the TAR-H model, but another model designated TAR-L is also produced. This is lighter in construction, faster in operation and is designed for the production of relatively smaller components, particularly in aluminium and other non-ferrous metals and alloys.

In Britain these tools are handled, with full technical service, by Tarex (England) Ltd., 22 Buckingham Gate, London, S.W.1.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

JANUARY

Birmingham

Tuesday, 22nd January, 6.30 p.m., in the James Watt Memorial Institute, Great Charles Street, Birmingham. Informal talk: "Stop Tail and Direction Indicator Lamps—Experimental Work to Establish Photometric Standards," by J. H. Nelson, Ph.D., B.Sc. (Associate Member).

Derby

Monday, 28th January, 6.15 p.m., in the Rolls-Royce Welfare Hall, Nightingale Road, Derby. Joint Meeting with the East Midlands Branch. Paper: "Quality in Engineering," by F. Nixon, B.Sc. (Member).

North-Eastern

Wednesday, 16th January, 7.30 p.m., in the Chemistry Lecture Theatre, The University, Leeds. Address by the Chairman of the Automobile Division, A. G. Booth, M.B.E. (Member), entitled "Experiences During Forty Years of Automobile Design."

North-Western

Tuesday, 22nd January, 7.15 p.m., in the Liverpool Corporation Passenger Transport Department Offices. Paper: "Jet Engines: Some Aspects of their Design, Operation and Manufacture," by A. W. Vickers, M.Sc.Tech. (Member).

Scottish

Monday, 21st January, 7.30 p.m., in the Institute of Engineers and Shipbuilders, 39 Elmbank Crescent, Glasgow, C.2. Paper: "Aspects in the Development of the Medium Weight Commercial Vehicle," by E. B. Stead (Associate Member).

Western

Thursday, 31st January, 6.45 p.m., in the Royal Hotel, Bristol. Series of short papers: "Servicing of Injectors," by R. Lewis, "The End of a Piston's Life," by A. C. Whitehorn (Associate Member), "Brakes," by J. G. Remington (Associate Member).

FEBRUARY

London

Tuesday, 12th February, 6.0 p.m., at 1 Birdcage Walk, Westminster, S.W.1. General Meeting. Series of short papers on engine detail components: "Sealing Rings for Wet Cylinder Liners," by J. L. Hepworth (Member), "Recent Developments and Modern Techniques in the Use of Gaskets," by M. G. Herrington, "The Design and Application of Thermostats to the Internal Combustion Engine Cooling System," by S. H. Blazey (Member), and "Gudgeon Pin Location," by R. C. Cross (Member).

Coventry

Tuesday, 5th February, 7.15 p.m., in the Grosvenor Room, Leofric Hotel, Coventry. Paper: "A Review of Hydrokinetic Fluid Drives and their Possibilities for the British Motor Industry," by J. G. Giles (Associate Member).

Luton

Monday, 11th February, 7.30 p.m., in the Assembly Room, Luton Town Hall. Paper: "Rubber Springs for Vehicle Suspension," by A. E. Moulton, M.A., and P. W. Turner, M.A., B.Sc.

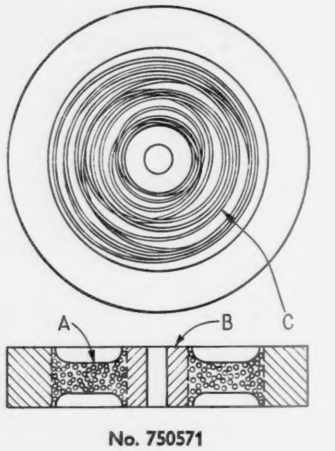
CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Reinforced plastics gear blanks

This invention relates to the manufacture of gears or gear blanks by compressing an artificial resin with a fibre insert, and is particularly concerned with the construction of the web portion between the hub and the rim. Various methods have been employed to form webs of material that could be treated with a binding agent and connected by pressure with a preformed rim, but these have proved to be relatively difficult, expensive, or lacking in strength.

In the method proposed the fibre insert of web portion A of the blank is built up about the hub portion B by a continuous loose winding and simultaneous shuttling of threads or strings. The winding may be performed in layers but the best results as regards strength are obtained if the shuttling movement is irregular and a disorderly wound coil C is obtained. In this way the threads cross one another and their interlocking during the pressing operation is improved.



No. 750571

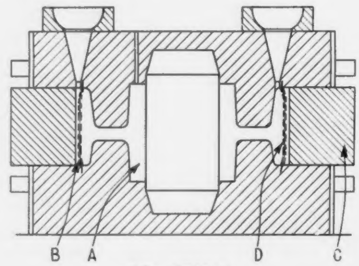
The preformed parts are hardened and consolidated under heat and pressure in a mould. Phenol-formaldehyde resin is preferable as the binding agent. *Patent No.* 750571, *Licentia Patent-Verwaltungs-G.m.b.H.* (*Germany*).

Composite cast articles

Certain components, which conveniently are of cast construction, may be subjected to localized wear. It may be difficult or uneconomical to incorporate areas of high-grade, wear-resistant material by the usual methods of fabrication and this method of simultaneous casting of two metals or alloys is proposed.

The example shows a mould for a roller requiring a hard metal rim, but the method may be used for any component; cams or wheels, for instance. The roller A is of a commercial grade of grey iron while the rim B is of white iron and subjected to a chilling action by the heavy metal ring C. A fusible ring D is set in the mould cavity, dividing it into two chambers each having a separate pouring connection.

Pouring of the two metals is effected simultaneously, the respective flows being appropriate to ensure that the levels of the two metals rise at substantially the same rate. The heat of the metals will



No. 750216

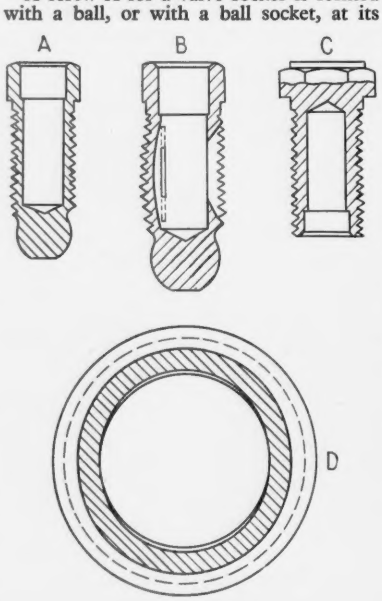
cause ring D to fuse, but such fusing is delayed until the pouring is completed. Thereafter, the two metals will intermix and coalesce and provide a homogeneous bond.

It is, of course, not essential that the mould should include a chill. The fusible ring may be of any metal or alloy that, preferably, will aid the welding action of the two metals, and the section is selected so that complete fusion is not effected prior to the filling of the mould. A sheet copper ring of 20 S.W.G. is cited as an example. Alternatively, a ring may be of preformed material chequered with depressions of relatively thinner gauge or a ring may be built up of two perforated sheets with an intermediate solid sheet. *Patent No.* 750216. *Millars' Machinery Co. Ltd.*

Self-locking tappet screw

Valve gear adjustment is commonly effected by manipulation of a screw, threaded in a tappet, push-rod, or rocker and secured by a lock-nut. With that construction, adjustment of valve clearance is awkward and time-consuming, as two tools and a feeler gauge are required. Accordingly, the invention provides a screw that is self-locking when fully engaged and needs only a single wrench for adjustment.

A screw A for a valve rocker is formed with a ball, or with a ball socket, at its

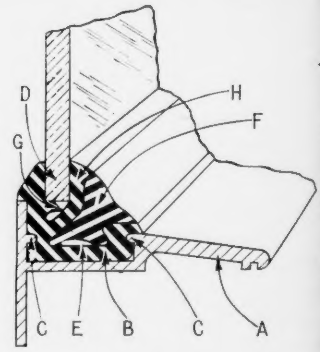


No. 749310

lower end and a head shaped to receive a wrench at its upper end. It is drilled axially from the upper end, and externally it is screwthreaded by any convenient method. A tool of the desired profile is inserted in the bore and the threaded wall is expanded at two or more circumferential locations, as indicated in the larger-scale section D. Owing to the shape of the tool and the stiffness imparted to the screw body by the ball end and the head, the expansion is limited to the intermediate threaded portion. The lower convolutions of the thread retain the original pitch diameter for easy entry into the threaded hole in the rocker. The screw body is heat-treated to a spring temper and the ball end may be hardened.

A modified design, shown at B, is slotted longitudinally at three equally-spaced circumferential locations between three expanded segments of the threaded body. The slots facilitate the expansion of the segments and enhance their resiliency after treatment.

At C is a form of screw suitable for use in a tappet body. This has a hardened contact element embodied with the head



No. 750252

at the upper end. The body, in this instance, is bored from the lower end and the entrance is counterbored to prevent its distortion by the expander tool. *Patent No.* 749310. *National Machine Products Company* (*U.S.A.*).

Glazing strip

The outstanding feature of this two-part glazing strip is the facility it provides for easy mounting of the window panel and also for its easy removal in the event of breakage. A flanged channel frame A is fitted in the window aperture and accommodates a main rubber strip B which is retained by ribs C. Throughout the length of the strip is a deep glazing channel D and alongside is a dovetailed channel E to receive the filler strip F. The base of the glazing channel is formed by a rib G and the relatively thin web H between the two channels springs from the supported wall of the main strip B below this rib. It is thus rendered more flexible and can be deflected through substantially 90 deg to permit entry or removal of the glass panel. The insertion of the filler strip F forces web H into tight engagement with the glass panel. *Patent No.* 750252. *Weathershields of Worcester, Ltd.*

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